

**AN EXPLOSION SUPPRESSION SYSTEM FOR
RAIL TANK CARS AND OTHER CONTAINERS
HOLDING PRESSURIZED LIQUIDS**

MECHANICAL DESIGN STUDY

TP 7986E

Prepared For:

**TRANSPORTATION DEVELOPMENT CENTRE
Montreal, Quebec**

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Transport
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Development Développement

REPORT DOCUMENTATION FORM

1 Transport Canada Report No. TP 7986E	2. TDC Project No. 6475	3 Recipient's Catalogue No.
4 Title and Subtitle An Explosion Suppression System for Rail Tank Cars and Other Containers Holding Pressurized Liquids - Mechanical Design Study	5 Report Date May 1986	6 Performing Organization Report No. C85-159
7 Author(s) A.M. Birk, C. Critch, D.E.J. Lamb, P.J. Turyk	8 Transport Canada File No. D1465-302-9	
9 Performing Organization Name and Address Davis Engineering Limited 1260 Old Innes Road Ottawa, Ontario K1B 3V3	10 DSS File No. 11SD.T8200-5-5524	11 DSS or Transport Canada Contract No. 1SD85-00231
12 Sponsoring Agency Name and Address Transportation Development Centre (TDC) Guy Favreau Complex 200 Dorchester Blvd. West, Suite 601, West Tower Montreal, Quebec H2Z 1X4	13 Type of Report and Period Covered Technical	14 Sponsoring Agency Code TDC 6475
15 Supplementary Notes Partially funded under the Energy R&D Program and Dept. of Supply & Services Unsolicited Proposals Program	16 TDC Project Officer D.W. Dibble	
17 Abstract A mechanical feasibility study for an internal thermal protection device (ITPD) for a rail tank car has been completed, covering structural design, operations and fabrication. Based on the structural analysis, an inner shell thickness and a layout for the support spacers were determined. The analysis included sloshing of lading. The presence of the internal shell would most significantly affect operations which were infrequently scheduled. The installation of an ITPD could be carried out for both new and old tank cars. The study showed that from structural and operations perspectives, the internal thermal protection device is feasible for pressurized tank cars. Structurally, the device can be practically designed to withstand the rigours of day to day operations and there do not seem to be any major unresolvable problems in the maintenance and operations of the car.		
18 Key Words Tank-car, explosion suppression, thermal protection, mechanical design, sloshing, operations	19. Distribution Statement A limited number of copies are available upon request from the Transportation Development Centre	
20. Security Classification (of this report) Unclassified	21. Security Classification (of this page) Unclassified	22. No. of Pages ---
23. Price ---		

Transports
CanadaTransport
Canada

Développement Développement

FORMULE DOCUMENTAIRE DU RAPPORT

1 N° du rapport Transports Canada TP 7986E	2 N° de l'étude — CDT 6475	3 N° de catalogue du destinataire
4 Titre et sous-titre An Explosion Suppression System for Rail Tank Cars and Other Containers Holding Pressurized Liquids — Mechanical Design Study		5 Date du rapport mai 1986
		6 N° du rapport de l'organisme exécutant C85-159
7 Auteur(s) A.M. Birk, C. Critch, D.E.J. Lamb, P.J. Turyk		8 N° de dossier Transports Canada D1465-302-9
9 Nom et adresse de l'organisme exécutant Davis Engineering Limited 1260 Old Innes Road Ottawa, Ontario K1B 3V3		10 N° de dossier — ASC 11SD.T8200-5-5524
		11 N° de contrat — ASC ou Transports Canada 1SD85-00231
12 Nom et adresse de l'organisme parrain Centre de développement des transports (CDT) Complexe Guy-Favreau 200 ouest, boul. Dorchester, Suite 601, Tour Ouest Montréal, Québec H2Z 1X4		13 Genre de rapport et période visée Technique
		14 Code de l'organisme parrain CDT 6475
15 Remarques additionnelles Partiellement financé par le programme énergétique et le programme des proposition spontanées du ministère des approvisionnements et services		16 Agent de projet — CDT D.W. Dibble
17 Résumé L'étude de faisabilité d'un dispositif interne de protection thermique pour wagons-citernes est terminée. Cette étude a porté sur les trois aspects suivants: structure, fonctionnement et fabrication. Une analyse de structure a permis d'établir le dimensionnement de la paroi interne et la disposition des entre-toises. Le ballotement de la cargaison a également été pris en considération. La présence d'une double paroi nuira certainement à l'exécution des visites à intervalle espacé. Le dispositif pourrait être installé tant dans les wagons neufs que dans les wagons usagés. L'étude a révélé que, du point de vue structurel et opérationnel, le dispositif de protection convient également aux wagons dont la cargaison est sous pression, le dispositif pouvant être dimensionné pour résister aux sollicitations de service et ne présentant aucune difficulté insoluble pour ce qui est de l'entretien et de l'exploitation.		
18 Mots-clés Wagon-citerne, anti-déflagration, protection thermique, conception mécanique, ballotement, exploitation		19 Diffusion Le Centre de développement des transports dispose d'un nombre limité d'exemplaires qui seront fournis sur demande.
20 Classification de sécurité (de ce rapport) non classifié	21 Classification de sécurité (de cette page) non classifiée	22 Nombre de pages —
23 Prix —		

ACKNOWLEDGEMENTS

Davis Engineering wishes to acknowledge the following individuals who assisted the authors through numerous discussions of the various aspects of this work: Mr. W. Watson of the NRC Railway Laboratory, Mr. L. Olson of the AAR, Dr. J. Sicilian of Flow Science Inc., and Dr. D. Kana of SwRI.

Davis Engineering also wishes to acknowledge the significant contributions of Mr. C. Comiskey of Procor Ltd. who acted as a consultant during the mechanical design study.

Funding by the Energy R&D Program and the Department of Supply and Services Unsolicited Proposals Program for this portion of the project is also acknowledged.

SUMMARY

The mechanical design study for the feasibility of an internal thermal protection device for a rail tank car proceeded in three parts, covering a structural design analysis, an operations study, and a fabrication study.

The structural analysis was undertaken using a finite element stress analysis computer code. A variety of loads typically encountered during tank car operation were applied to the model. Based on the analysis results an inner shell thickness and a layout for the support spacers were determined. Table 1.3-1 displays the principal results of the structural analysis.

It was determined early in the study that loads due to longitudinal sloshing of the lading was the determining load case for the design of the internal shell and its support scheme. For this reason, considerable effort was made to determine accurate slosh load figures.

The sloshing analysis was performed with the aid of a two dimensional finite difference computer code for the motion of fluids with free surfaces. After an appropriate solution domain and solution approach were determined for the system, the code was used to calculate fluid pressures exerted on the walls of the internal shell and external tank for several fill conditions.

Because the code was two dimensional, the calculations could only be used in a limited and conservative manner. However, the magnitude of the pressures were not unreasonable. A lack of sufficient experimental evidence from the open literature prevented any qualitative assessment of the calculated sloshing results.

The maintenance and operations section reviewed existing procedures regarding tank loading and unloading, in-transit operations, changing commodities, cleaning, inspection and repairs. The impacts on these procedures caused by the internal protection device were examined, and possible changes to accommodate the internal shell were proposed. Table 1.3-2 presents a summary of the results of the maintenance and operations study.

It was determined that the presence of the internal shell would most significantly affect operations which were infrequently scheduled, such as thorough inspections. In addition, changes to existing more frequent procedures would be required for internal shell equipped cars.

From a fabrication standpoint, the internal shell can be installed through one end of a new or old tank car. For refitting it will be necessary to remove one end-cap. Because the internal shell is made of the same material as the rest of the car there are no anticipated problems associated with heat treatment, welding, or corrosion.

The study showed that from structural and operations perspectives, the internal thermal protection device is feasible for pressurized tank cars. Structurally, the device can be practically designed to withstand the rigours of day to day operations and there do not seem to be any major unresolvable problems in the maintenance and operations of the car.

It is recommended that further study be undertaken to assess the thermal protection and fluid dynamic aspects of the device.

Description	Recommended Value
Inner Shell Thickness	3mm (1/8 inch)
Inner Shell Weight	47.2 kN (10600 lbs)
Axial Support Spacing (Along Slot)	0.3 m (1 ft)
Axial Support Spacing (Elsewhere)	1.2 m (4 ft)
Radial Support Spacing	30°
Support Size (Along Slot)	1 1/2" dia bars
Support Size (Elsewhere)	1 1/4" x 1/4" tubing

Table 1.3-1 Structural Analysis Summary

Item	Impact	Quantity	Solution/Comment
Loading/Unloading	- Valve Train Interference		- Provide passage through inner shell for valve train
	- Differential Pressure Buildup (Vapour Bubble in inner cavity)		- Provide manual vent valve between inner cavity and annulus
Payload	- Weight Capacity	- 6% reduction for high filling density commodity	- not applicable here
	- Volume Capacity	- 0.5% reduction for LPG and anhydrous ammonia	
In Transit Operation	- identification placard		- include placard identifying that the tank is internal shell equipped
	- changes in coupler/braking loads	- 6% increase for unloaded car	- not expected to be significant
Commodity Change	- Internal tank cleaning	- 200% more internal surface area to clean	- longer cleaning time; test to determine how much longer
	- manway access for internal mop up		- include inner shell hatch or incorporate it with outer shell hatch
Inspections	- inability to inspect annulus between internal shell and external tank for corrosion	- detailed inspections required every 10 years	- infrared camera analysis followed by fiber optic analysis
			- ie. use other non-destructive inspection techniques
			- semi yearly inspections after commodity change
			- access to critical areas facilitated by a limited number of removable access ports
Repair Procedures	- accessibility to damaged pressure vessel region from interior		- cut away inner shell in the damaged pressure vessel area and patch after repair is complete
Fabrication	- structural sagging of pressure vessel during heat treating		- pipes tack welded to structure for support of main shell (same technique as presently used)
	- oxide formation on inner shell during heat treating		- oxide flushed away sufficiently during hydrostatic testing

Table 1.3-2 Maintenance and Operations Summary

Item	Impact	Quantity	Solution/Comment
Refit	- insertion of inner shell into existing tank cars		- cut off elliptic end of external pressure vessel and insert inner shell
	- cost	- \$36,000 for prototype - \$25,000 each for production	- one time cost
Corrosion	- rate of corrosion of inner shell	- negligible reduction of wall thickness over 20 years	- corrosion allowance for inner shell likely is not required; testing necessary to confirm this
Government Regulations	- specifications approval of Canadian Transportation Commission and Association of American Railways		- must demonstrate safety advantage of internal thermal protection device equipped tank cars

Table 1.3-2 Maintenance and Operations Summary

SOMMAIRE

L'étude de faisabilité d'un dispositif interne de protection thermique pour wagons-citernes a porté, côté mécanique, sur les trois aspects suivants: structure, fonctionnement et fabrication.

L'analyse de la structure s'est faite par la méthode des éléments finis assistée par ordinateur, utilisant un programme qui permet de prendre en compte les charges types s'exerçant normalement sur un wagon-citerne. L'étude a débouché sur le dimensionnement de la paroi intérieure et sur la disposition des entretoises. Ces caractéristiques sont récapitulées dans le tableau S.1.

Il avait été déterminé dès le début de l'étude que le principal paramètre pour le dimensionnement de la paroi intérieure et de son système d'appui étaient les charges longitudinales dues au ballotement de la cargaison. C'est pourquoi, les valeurs correspondantes ont été soigneusement déterminées.

Un code informatique bidimensionnel a été utilisé pour l'analyse des charges par la méthode des différences finies, à l'égard d'un fluide à surface libre. Après avoir déterminé un domaine et une méthode de résolution appropriés, ce code a permis de calculer la pression du fluide sur l'extrémité des parois intérieure et extérieure pour différents niveaux de remplissage.

Mais, étant donné que le code était bidimensionnel, le résultat des calculs sur les déplacements des liquides a dû être interprété de manière restrictive et dans le sens de la prudence. Néanmoins, l'amplitude des pressions a été trouvée raisonnable. Quant au résultat des calculs sur les charges dues au ballotement, il n'a pas été possible de procéder à une évaluation qualitative à cause de l'insuffisance des informations empiriques publiées.

Côté entretien et fonctionnement, l'étude a fait un tour d'horizon des procédures courantes de chargement/déchargement des wagons-citernes, de vérification en cours de route, de nettoyage en vue d'une substitution des cargaisons, de visite et de réparation. L'influence du dispositif de protection sur ces procédures a été étudiée, et des changements ont été proposés dans l'éventualité de la mise en place du dispositif. Toutes les recommandations concernant cet aspect de l'étude sont reprises dans le tableau S.2.

L'étude a montré que les visites à intervalle très espacé comme les visites détaillées, auront le plus à souffrir de la présence du dispositif et que des modifications seront, en outre, nécessaires en ce qui concerne les procédures des visites planifiées, plus fréquentes.

Du point de vue de la fabrication, la technique à utiliser serait de faire entrer le dispositif par une extrémité, dans le cas d'un wagon neuf. En rééquipement, il faudra enlever une des parois d'extrémité du wagon. Côté traitement thermique, soudage ou corrosion, aucun problème n'est prévu étant donné que le dispositif est fait du même matériau que la citerne.

L'étude a révélé que, du point de vue structurel et opérationnel, le dispositif de protection convient également aux wagons transportant une cargaison sous pression, le dispositif pouvant être dimensionné pour résister aux sollicitations de service et ne présentant aucune difficulté insoluble pour ce qui est de l'entretien et de l'exploitation.

Le rapport recommande de poursuivre l'étude par la recherche sur la réponse du dispositif de protection au comportement dynamique de la cargaison.

DESCRIPTION	VALEUR RECOMMANDEE
Epaisseur de la paroi intérieure	3 mm (1/8")
Poids de la paroi intérieure	47,2 kN (10600 lb)
Espacement des entretoises longitudinales (le long de l'ouverture)	0,3 m (1 pi.)
Espacement des entretoises longitudinales (ailleurs)	1,2 m (4 pi.)
Angle des entretoises radiales	30°
Diamètre des entretoises (le long de l'ouverture)	barres 1½"
Diamètre des entretoises (ailleurs)	tubes 1¼" x 1¼"

Tableau S-1 Caractéristiques du dispositif de protection thermique

Elément	Influence	Valeur quantitative	Solution/commentaire
Chargement/déchargement	Présence gênante de la raccorderie de sécurité		Pratiquer dans la paroi intérieure un passage suffisant pour la raccorderie et la rampe des soupapes.
	Pression différentielle (bulle gazeuse dans la citerne)		Prévoir un clapet manuel de mise à l'air libre pour égaliser la pression des deux côtés de la paroi intérieure.
Réduction de la charge marchande	En poids	6 p. 100 dans le cas d'une cargaison à densité de remplissage élevée	- Ne s'applique pas ici
	En volume	1/2 p. 100 dans le cas du GPL et de l'ammoniac anhydre	
Vérification en cours de route	Placard-étiquette		Prévoir un placard-étiquette indiquant la présence d'un dispositif de protection
	Variation dans les efforts à l'attelage et en freinage	6 p. 100 de plus en marche à vide	Aucune influence significative
Substitution de cargaison	Nettoyement intérieur	Surface à nettoyer augmentée de 200 p. 100	Procéder à des essais pour déterminer le temps supplémentaire à consacrer au nettoyage
	Trou d'homme pour parvenir à l'intérieur de la citerne		Prévoir une ouverture dans la paroi intérieure ou l'intégrer au trou d'homme existant.

Tableau S.2 Recommandations concernant l'entretien et le fonctionnement

(suite)

Elément	Influence	Valeur quantitative	Solution/commentaire
Visites	Impossibilité de visiter l'espace annulaire pour déceler la présence de corrosion	Visite détaillée un fois par 10 ans	Contrôle à l'infrarouge suivi d'un contrôle par une méthode optoélectronique Toute autre méthode de contrôle non destructif Visites semi-annuelles après une substitution de cargaison Prévoir des ouvertures donnant accès aux endroits les plus critiques.
Réparations	Accès intérieur aux parties endommagées de la citerne		Couper la paroi intérieure au droit de ces parties et faire le rapicage par la suite.
Fabrication	Déformation du wagon due au traitement thermique Formation d'un dépôt d'oxyde durant le traitement thermique		Utiliser le procédé de soudage par points pour soutenir la structure, comme cela se fait couramment L'eau servant aux essais hydrostatiques assurera un lessivage suffisant.
Rééquipement	Introduction du dispositif de protection dans un wagon Coût		Enlever une des extrémités elliptiques du wagon et introduire le dispositif. Déboursé exceptionnel
Corrosion	Taux de corrosion de la paroi intérieure	Amincissement négligeable sur 20 ans	Il ne faudra probablement pas tenir compte du facteur corrosion, mais des essais devront être faits pour le confirmer.
Réglementation	Approbation du descriptif par la Commission canadienne des transports et l'Association of American Railroads		L'utilité du dispositif de protection devra être démontrée.

Tableau S.2 Recommandations concernant l'entretien et le fonctionnement

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A Inner Shell Fatigue Analysis

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1.0 INTRODUCTION

This document is the milestone report describing Task 1, the Mechanical Design Study of Phase I, of the Tank Explosion Suppression Project. The work was performed under DSS Contract 11SD.T8200-5-5524 for the Transportation Development Centre of Transport Canada. The contract was a result of an unsolicited proposal submitted by Dr. A.M. Birk of Davis Engineering Ltd.

The submission of this report marks a go/no go decision point in the Tank Explosion Suppression Project. A go decision would mean the continuation on to Task 2, the Thermal Design Study.

The work contained herein was performed exclusively by Davis Engineering Limited, Ottawa, Ontario.

1.1 BACKGROUND

When a tank car (or any other container holding a pressurized liquid) is exposed to external fire impingement, there is a risk of thermally induced rupture. Such a rupture is usually caused by reductions in the strength of the tank primary shell due to the development of high wall temperatures where the wall is wetted only by vapour. In all other wall areas wetted by liquid, the temperatures remain relatively low due to the effective cooling by the liquid.

Presently, tank cars are protected against thermal induced ruptures by the use of pressure relief valves and thermal insulation. These devices have proven to be extremely effective provided that they have been properly maintained and protected from any damage (ie because insulation is applied externally it can be damaged during an accident, and its effectiveness can be significantly reduced). Therefore, research must continue to ensure that tank cars or other containers for transporting or storing hazardous commodities are protected using the best available means. This study involves a mechanical feasibility study of a new protection concept that could potentially further reduce the risk of thermal induced rupture.

The device consists of an internal shell which is supported within a tank in such a way that an annular space exists between it and the tank wall. An opening along the bottom allows communication between the annulus and the inner cavity. Figure 1.1-1 shows a cross-sectional view through a tank car with the device installed.

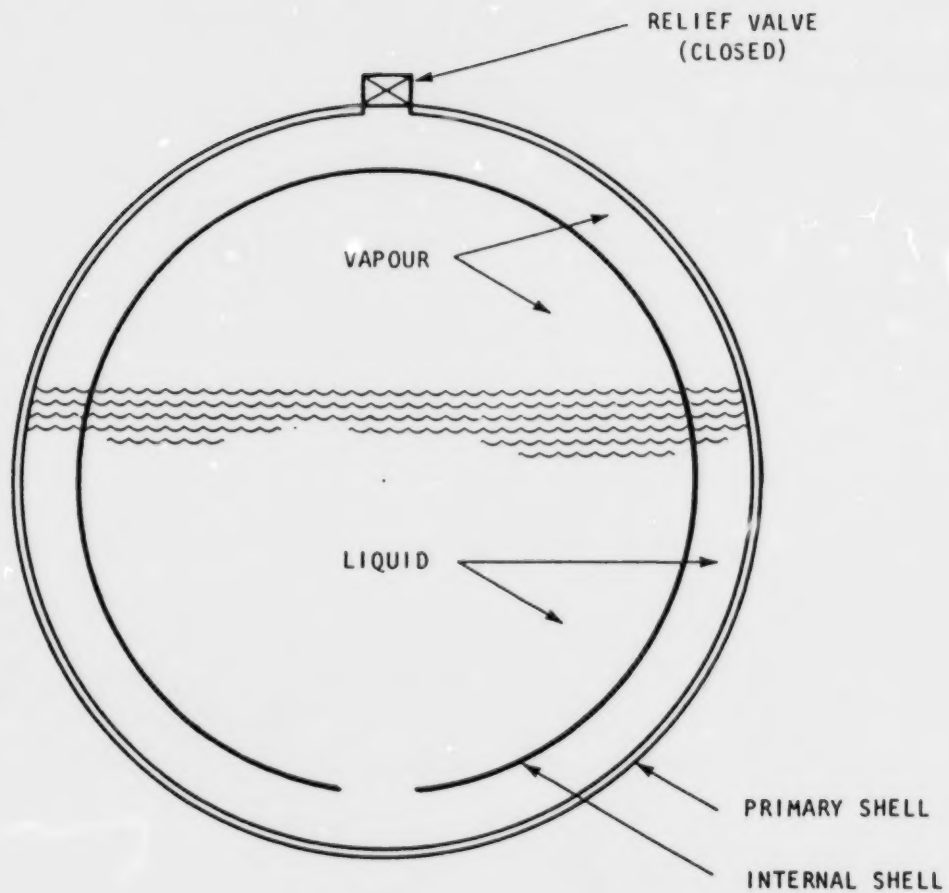


FIGURE 1.1-1 TANK WITH DEVICE WHEN RELIEF VALVE IS CLOSED

The figure shows the situation where the relief valve is closed and the device has no effect on the tank or its lading. It is when the relief valve is opened, that the device has its beneficial effects.

The device has two modes of operation. The first mode has the effect of maintaining low wall temperatures when the relief valve is open, the second mode results in vapour flow through the relief valve when the tank is rolled over on its side.

The first mode is in action when the tank is upright or slightly rolled (the actual roll angle is not known at this time and it is possible that the device will remain effective at large angles of roll). Figures 1.1-2 and 1.1-3 show the case where the relief valve is open and the device is acting in the first mode. When the relief valve is open and the device is present, the flow out of the relief valve causes a pressure drop between the inner cavity and the annular space. This pressure difference "sucks" the liquid up to a higher level in the annular space. This immediately reduces the vapour wetted area. In addition, the restricted flow area in the annulus results in very high flow velocities which entrains liquid droplets. This high velocity, wet two-phase flow results in relatively high heat transfer coefficients with the net effect of reducing the wall temperatures at the top of the tank.

The second mode of the device operation comes into play when the tank roll is large. Under normal conditions (tank without device) a tank that is rolled will have its relief valve submerged in liquid. If the relief valve is properly sized for vapour flow, it may not be adequate for liquid flow. Therefore it is desirable that the valve always pass vapour or dry two-phase flows.

The second mode of the device operation ensures vapour flow when a horizontal or near horizontal tank has a large roll angle. Figure 1.1-4 shows a tank rolled to the probably maximum (120°) angle. When the valve first opens it will vent liquid until the liquid level in the inner cavity is below the opening in the inner shell. When the annulus is clear of liquid, then the valve will pass vapour or dry two-phase fluids only thus ensuring efficient pressure relief.

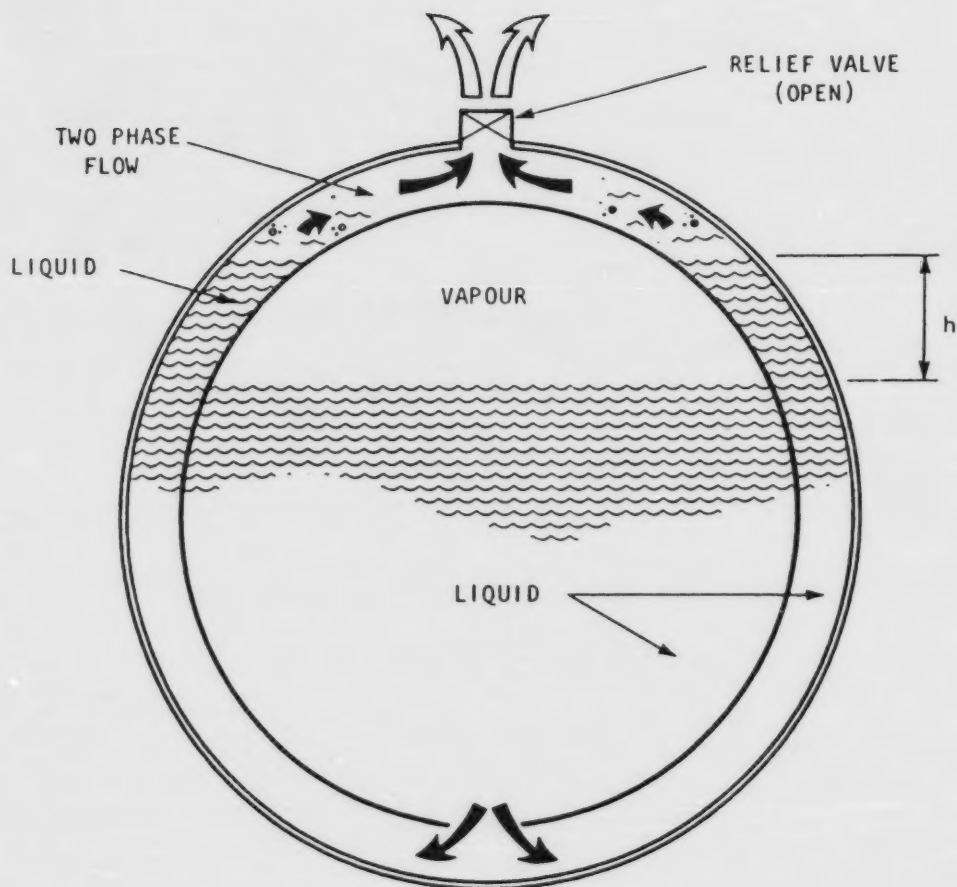


FIGURE 1.1-2 TANK WITH DEVICE WHEN RELIEF VALVE IS OPEN

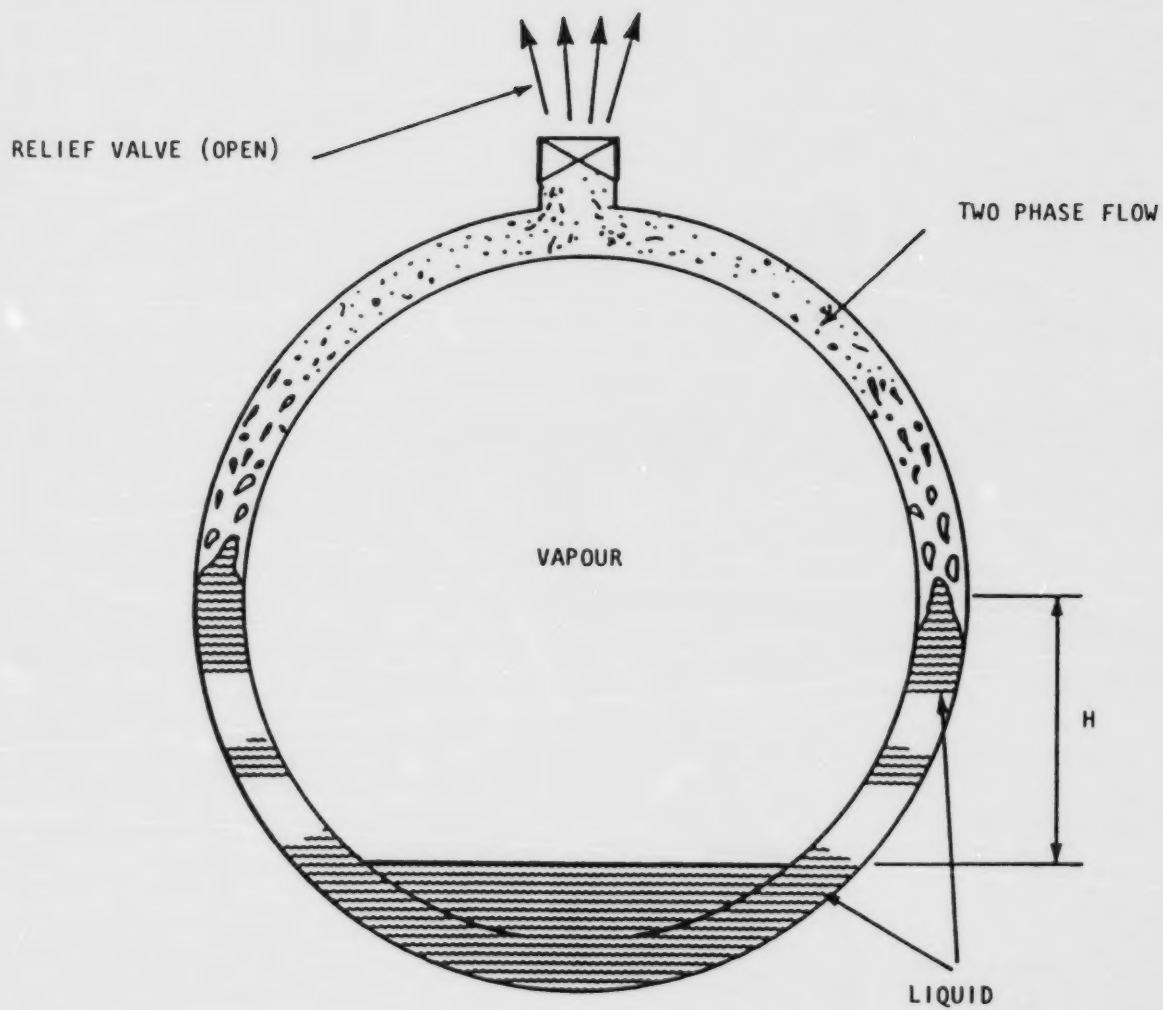


FIGURE 1.1-3 TANK WITH DEVICE WHEN RELIEF VALVE IS OPEN
(TANK ALMOST EMPTY)

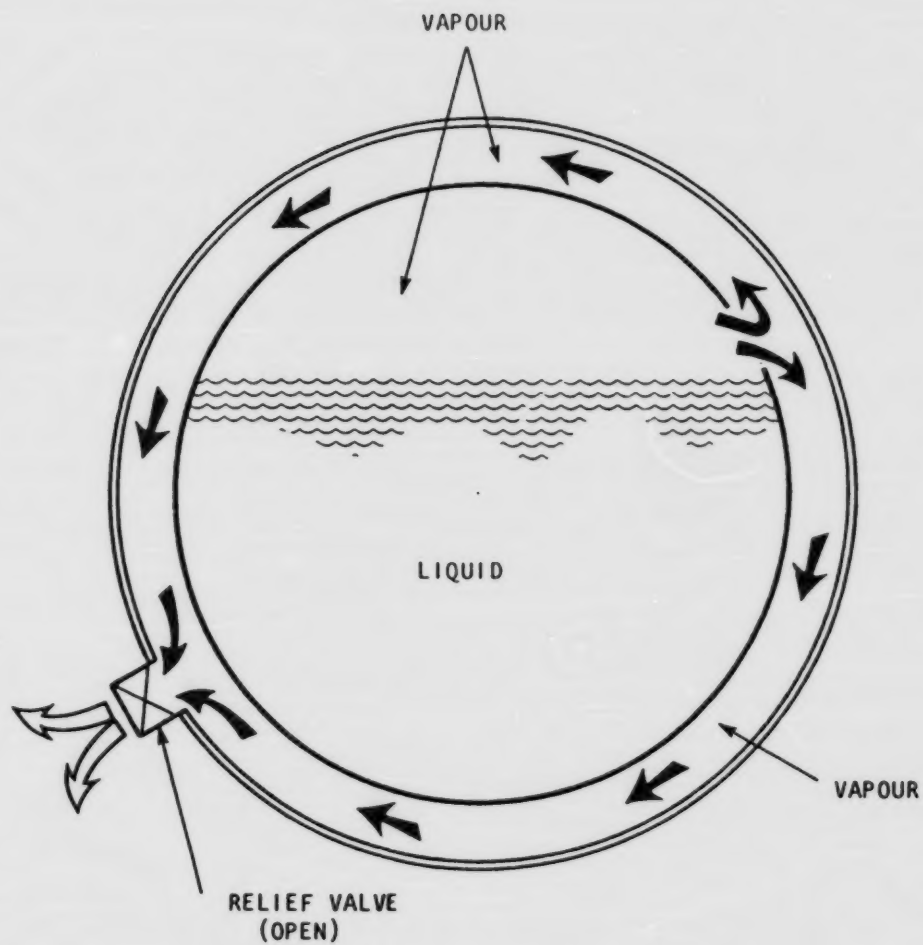


FIGURE 1.1-4 OVERTURNED TANK WITH DEVICE WHEN RELIEF VALVE IS OPEN

1.2 OBJECTIVES AND SCOPE

The primary objective was to assess the feasibility of using an internal shell for thermal protection in tank cars from structural and operational viewpoints, and to provide a preliminary mechanical design scheme in conjunction with this.

The study scope was limited to a preliminary evaluation of the various design, operation, and fabrication considerations that determine whether or not the concept is feasible. The study was to consider:

- structural design, including;
 - loading environment (sloshing, operational accelerations, vibration)
 - shell thickness and support means
- operational considerations, including;
 - cleaning
 - loading/unloading
 - inspection
 - corrosion
- fabrication considerations, including;
 - new car installation
 - retrofitting
 - heat treating
 - manufacturing methods
 - weights
 - estimate costs

The tank vehicle considered in this study is the jacketed pressurized rail tank car designated CTC-112J 340 W carrying LPG or anhydrous ammonia. The tank car is 18.3 m (60 ft.) long and 3.05 m (10 ft.) in diameter with elliptical heads, and has a capacity of 127 m³ (33,600 US gal).

2.0 PRELIMINARY STRUCTURAL DESIGN AND ANALYSIS

This section of the report presents in detail the approach taken and results obtained in the preliminary design of the inner shell. The tank car was modelled using a commercially available finite element package developed by Swanson Analysis Systems (ANSYS). Various design load cases, as detailed later in this section, were applied to the model. The inner shell thickness and support layout were determined based on the results of the finite element analysis.

2.1 TANK CAR STRUCTURAL MODEL

As stated above, the internal shell structure was analysed using finite element methods. The following sections describe, in detail, the structural tank car model.

2.1.1 Element Size

The first step in the model generation was to determine an element size that would accurately represent the structure while maintaining a reasonable solution time. To this end, a section of the inner shell was modelled with a hydrostatic pressure distribution corresponding to the pressure exerted by the lading in an inverted tank. The three different meshes shown in Figure 2.1.1-1 to Figure 2.1.1-3 were tested. The accuracy of the results was evaluated assuming the results obtained from the finest mesh (Figure 2.1.1-3) were the most accurate. Table 2.1.1-1 presents a comparison between stresses and displacements at two random points (A and B on Figure 2.1.1-1) in each mesh. Stresses are also presented in the spacer pin at 30° from the top. The stresses used for the shell are stress intensity values, defined as the difference between the maximum and minimum principal stresses.

The coarse mesh of Figure 2.1.1-1 did not represent the inner shell section accurately. The calculated stresses and displacements compared poorly with the model of Figure 2.1.1-3 as is illustrated in Table 2.1.1-1.

Much better results were obtained with the second model (Figure 2.1.1-2). It was found that the results were generally within 10% of the values obtained from the model of Figure 2.1.1-3. Based on this test it was concluded that the element size shown in Figure 2.1.1-2 was adequate in the modelling of the structure and would provide the optimum compromise between accuracy and solution time.

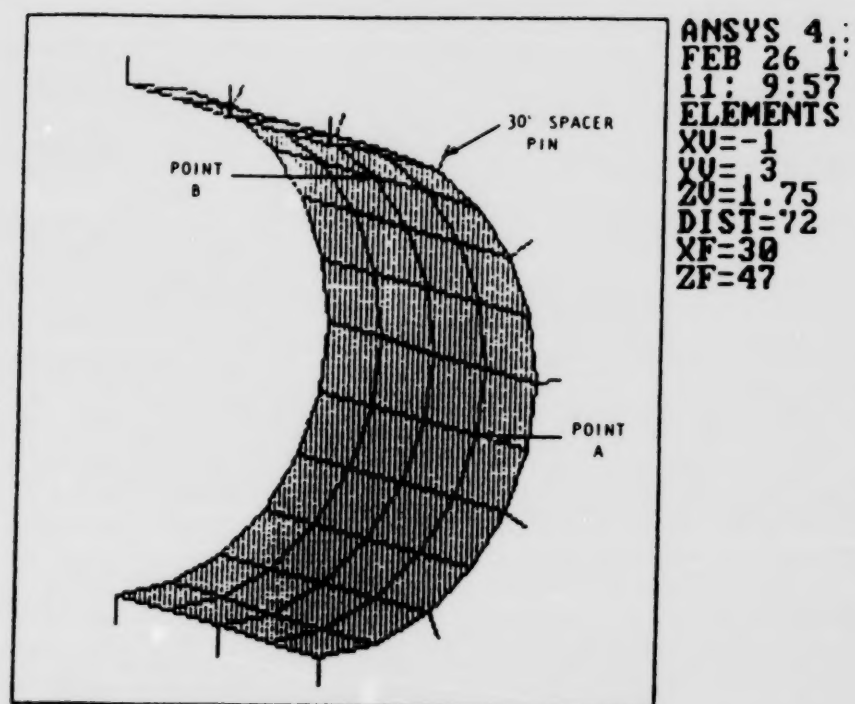
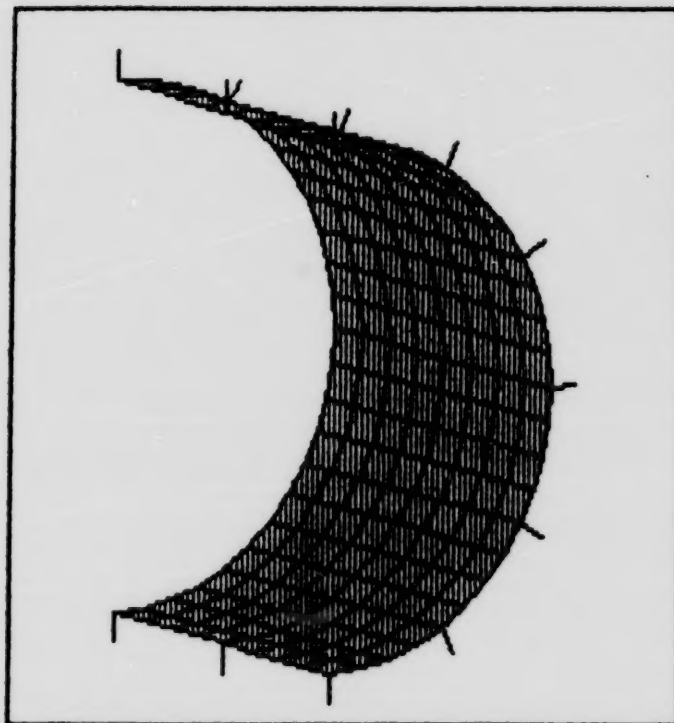
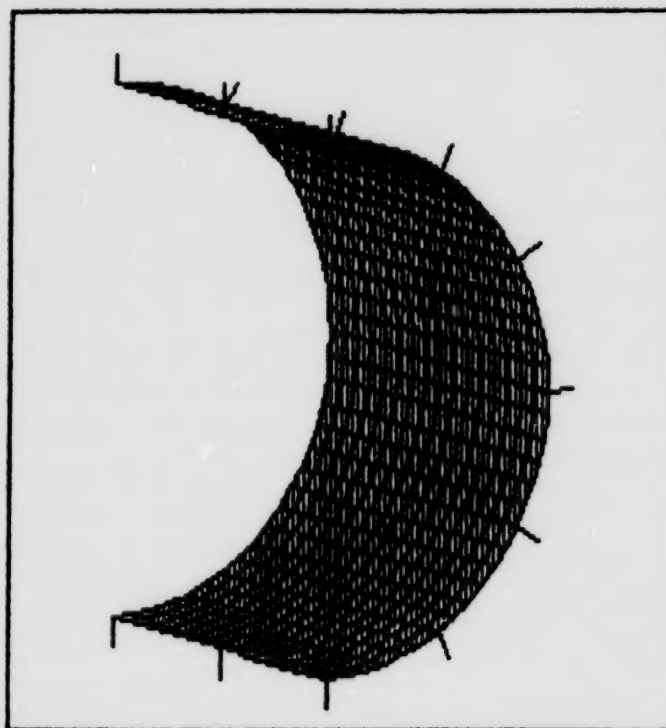


FIGURE 2.1.1-1 TEST FOR ELEMENT SIZE - MESH 1



ANSYS 4.1
FEB 26 11:24:44
ELEMENTS
XU=-1
YU=3
ZU=1.75
DIST=72
XF=30
ZF=47

FIGURE 2.1.1-2 TEST FOR ELEMENT SIZE - MESH 2



ANSYS 4.1
FEB 26 11:51:52
ELEMENTS
XU=-1
YU=3
ZU=1.75
DIST=72
XF=30
ZF=47

FIGURE 2.1.1-3 TEST FOR ELEMENT SIZE - MESH 3

		MESH 1	MESH 2	MESH 3
POINT A	S.I.* ($\sigma_1 - \sigma_2$) (MPa)	11.7	11.3	11.3
	% Difference	3.6%	0.4%	0%
	Vertical Disp. (MM)	-0.708	-0.816	-0.833
	% Difference	15%	2.1%	0%
POINT B	S.I.* ($\sigma_1 - \sigma_2$) (MPa)	4.30	2.80	2.54
	% Difference	69%	10%	0%
	Vertical Disp. (MM)	-0.565	-0.875	-0.952
	% Difference	40%	8%	0%
SPACER	$\sigma_{\max}$ (MPa)	126	127	128
	% Difference	1.1%	0.2%	0%

* Stress Intensity; σ = stress (MPa)

TABLE 2.1.1-1 COMPARISON OF STRESSES AND DISPLACEMENTS
FOR TEST MESHES

2.1.2 Model Assumptions and Analysis Approach

The tank car model was based on the dimensions of a CTC-112J 340 W tank car. A gap of 150 mm (6 in.) was assumed between the inner and outer shells. A ratio for the inner elliptical head of 2.25:1 was used to maintain the 150 mm gap at the tank ends. Mechanical properties for TC128-B steel were assumed for all structural components. Taking advantage of symmetry it was possible to model either 1/4 or 1/2 of the tank depending on the load case.

Due to the large number of elements required to model the inner shell accurately, and restrictions on model size imposed by computer limitations, it was not possible to analyze the entire tank car. Instead, the inner shell and spacer pins were modelled separately as shown in Figure 2.1.2-1. The spacer pins were assumed to be fixed rigidly in space at their base. The outer shell essentially acts as a rigid support since it is much thicker than the inner shell. In order to validate this simplification, the entire tank car was modelled (i.e., - both outer and inner shells) using a coarser mesh (Figure 2.1.2-2), and compared with a similar model of the inner shell only. These models were too coarse to use in the actual analysis, but made it possible to estimate the error introduced by modelling the inner shell alone. On comparison of the results from these models it was found that the stress results from the model of the inner shell by itself were within 10% to 15% of the values obtained from the inner/outer shell model. Based on these findings it was concluded that modelling the inner shell by itself would not introduce significant error.

The loads due to sloshing governed the design, hence were analyzed first. Using the model of Figure 2.1.2-2, an adequate spacer layout and inner shell thickness were found. The resulting configuration was then analyzed for the remaining load cases. The various load cases are described in the next section.

The final portion of the preliminary mechanical design was a modal analysis to determine the natural frequencies of the inner shell when the tank car is empty.

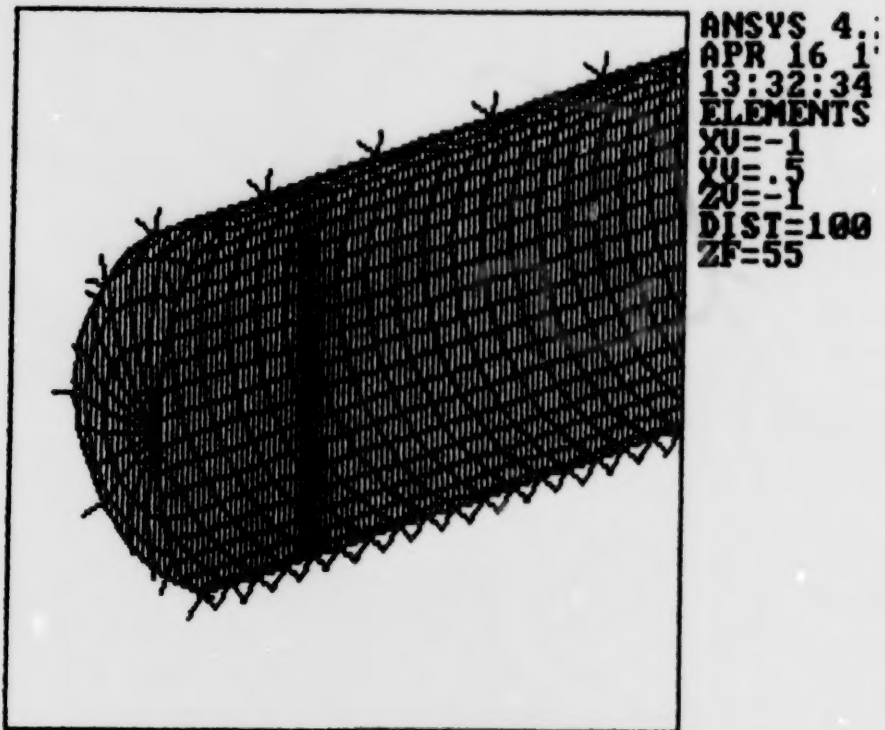


FIGURE 2.1.2-1 INNER SHELL AND SPACER PIN MODEL USED IN ANALYSIS

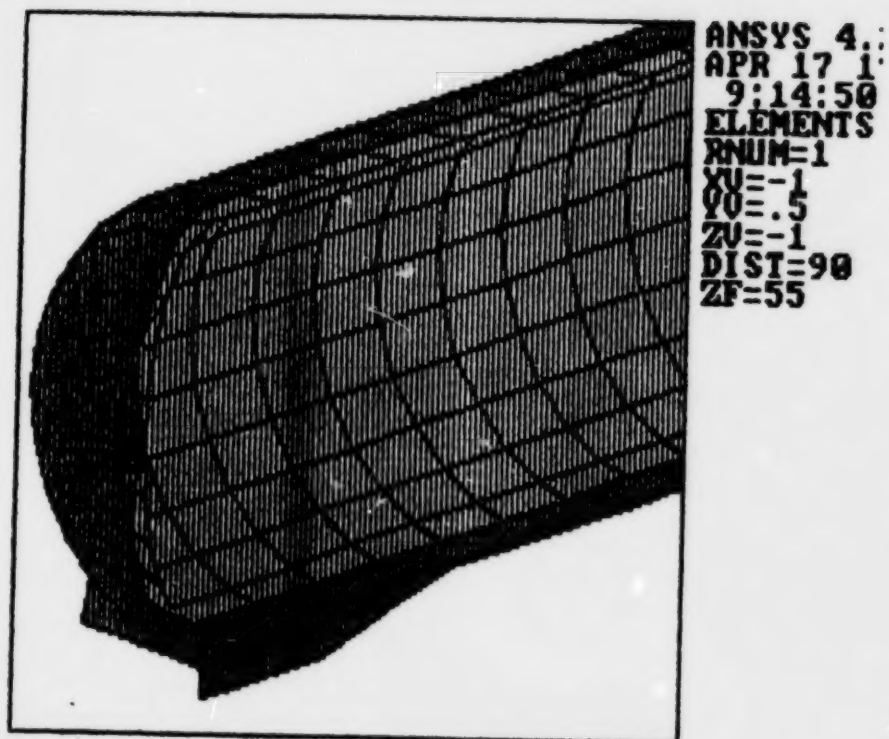


FIGURE 2.1.2-2 MODEL OF INNER AND OUTER SHELL (COARSE MESH)

2.2 DESIGN LOAD CASES

The inner shell of the tank car must be capable of withstanding a large variety of loads. The design of the inner shell was based on several load cases for both an empty and laden tank.

For an empty tank the maximum operational accelerations in the axial, lateral and vertical directions were considered. As well, a dynamic analysis was performed to determine the natural frequencies of an empty tank car with an inner shell. For a laden tank the inner shell must be designed to support the fluid weight if inverted, sloshing loads, and the pressure differential between the annulus and inner cavity when the relief valve is open. The internal shell must also be able to conform to the shape of the primary shell as it deflects under the vertical dynamic loads imposed by the lading. The various design cases are summarized in Table 2.2-1.

TANK FILL	ANALYSIS CASE
LADEN TANK	Sloshing Loads/ Operational Accelerations
	Fluid Weight (Inverted Tank)
	Differential Pressure (Valve Open)
	Primary Shell Deflections
EMPTY TANK	Operational Accelerations
	Natural Frequencies

TABLE 2.2-1 STRUCTURAL ANALYSIS CASES

2.2.1 Laden Tank

The loads for a laden tank car were analyzed first. These loads are more severe than the empty tank loads, hence, they will govern the inner shell design. This subsection discusses in detail the laden loads and any related assumptions.

2.2.1.1 Inverted Tank

It is possible that in an accident situation, a tank car may turn over. In this case the inner shell would be required to support the weight of the lading. This case was modelled as a hydrostatic pressure acting on the walls of the inner shell (including the elliptical heads).

For a 127 m³ (33,600 US Gal) CTC-112J340 W Tank Car, the maximum allowable payload is 780 kN (175,400 lb). The worst case for an inverted tank car occurs when the inner shell must support the entire 780 kN (175,400 lb) of lading. To model this case it was assumed the fluid completely fills the inner shell. Hence, the fluid density was calculated by dividing 780 kN by the entire volume contained within the inner shell. This is equivalent to the inner shell being filled by a fluid with a specific gravity of 0.77. This is conservative as it is unlikely that in a crash situation all of the fluid will be contained within the inner shell. Some fluid will initially be caught in the annular gap between shells.

2.2.1.2 Sloshing

Early on in this feasibility study it was concluded that the structural design of the internal shell would be dictated by axial slosh loading. For this reason, considerable effort was concentrated on defining realistic slosh loads in a tank-car. The slosh loading analysis cannot be adequately addressed here and therefore will be described in Section 2.3 to follow.

2.2.1.3 Pressure Differential

The final load case for a laden tank is a pressure differential between the annulus and inner cavity. This occurs when the tank is upright and the relief valve is open. In order to determine the pressure difference that will occur, it is necessary to perform a thermal/fluids analysis of the system. Since the thermal analysis is not part of the preliminary mechanical design, this load case cannot be addressed at this point in the project. However, it is not anticipated that the pressure differential will cause any problems, as it is unlikely to be as severe a condition as either of the two cases discussed previously.

2.2.1.4 Primary Shell Deflections

This load case considers the response of the inner shell to movements of the primary shell.

Since the primary shell is several times thicker than the inner shell, it will be several times less compliant under load. As such, during any static or shock load seen by the structure which causes deformation of the primary shell, the inner shell will move sympathetically and see a level of stress several times less than that of the primary shell. The transmission of deflections from the primary shell therefore, will not be a critical design case. It would, of course, be considered during the detailed design of the internal shell supports.

2.2.2 Empty Tank

To complete the mechanical design of the inner shell, two load cases for an empty tank were considered.

2.2.2.1 Operational Accelerations

The tank car must be capable of withstanding a variety of operational loads. The peak accelerations expected during the life of the tank car are: 12.0 g axial, 2 g lateral, and 3 g vertical. These loads were analyzed last as they are less severe than the previous load cases, and do not alter the inner shell design. The above accelerations were obtained from Reference (19).

2.2.2.2 Natural Frequency

Once the preliminary spacer layout and shell thickness were determined it was possible to perform a dynamic analysis to find the natural frequencies of a tank car with inner shell. This was done for an empty tank only as it was felt that any excitations in a full or partially full tank would be damped out by the lading.

2.3 SLOSHING LOADS

As stated earlier in this report, it is believed that the most severe loading on the internal shell will be caused by fluid sloshing. Because a liquid is able to move freely inside a container, an acceleration or deceleration of the container causes relative fluid motion, or sloshing. This in turn imparts a momentum impulse to the

container, which creates a motion stability problem for the tankers; and more importantly for the present purposes, it creates a pressure load on the surfaces inside the container.

Slosh loading on the interior structure of a tank car can occur several times per day in typical switchyard operations. Although the outer shell is more than adequately designed presently to withstand slosh loading, the inner shell must also be designed to do so.

This section addresses the problem of sloshing by predicting the pressures caused by the fluid motion. These pressures are used in the stress analysis (Section 2.4) to determine an adequate wall thickness and support configuration for the inner shell.

2.3.1 Problem Statement

The problem involves the determination of the pressures in excess of the normal operating pressures, which are caused by fluid motion or sloshing relative to the tank surfaces. The type of fluid motion in a tank is three dimensional and involves a free surface, which causes the motion to be very complex and extremely difficult to analyse using simple techniques. With the introduction of the internal shell, the problem becomes even more complex because the pressures built up in the annulus due to the flow in that region will tend to offset the pressures built up on the internal shell inner surface due to sloshing.

The analysis was intended to represent the most severe sloshing effects in a CTC-112J 340 W tank car. The lading was assumed to be LPG. The tank was 18.3 m long by 3.05 m in diameter. The internal shell was 18.0 m long by 2.74 m in diameter which gave a uniform gap of 0.15 m.

Although sloshing can result from accelerations in any direction relative to a tank car, it is the sloshing caused by severe accelerations along the longitudinal axis, such as those caused by humping or braking, that will likely cause the largest slosh forces. For this reason the analysis dealt only with longitudinal slosh effects. It was assumed that the most severe sloshing occurs during a humping mode of operation.

Railway laboratory simulations (1) of switchyard operations consist of decelerating a railcar in order to produce a 5.56 MN (1,250,000 lbf) load at the coupler. The maximum initial velocity considered is 6.3 m/s

(14 mph) if the 1,250,000 lbf coupler load is not achieved. The most severe slosh loading will occur when the initial fluid velocity is greatest, thus the latter criterion (14 mph initial velocity) in switchyard impact tests was considered in this analysis.

It was desired to compute the wall pressure distributions on the external tank and on the inner shell which results from the fluid motion. Several outage situations were considered. Because the motion of the fluid under these conditions is a transient (time-varying) phenomenon, it was necessary to determine the maximum pressures during the deceleration process.

2.3.2 Solution Method

The fluid motion within the tank was calculated with the computer code SOLA-VOF (Solution Algorithm - Volume of Fluid) which was developed at the Los Alamos Scientific Laboratory (2,3). SOLA-VOF solves the full time dependent two dimensional Navier-Stokes equations for viscous fluid flow. The unique feature of the program is its capability to handle systems with free liquid surfaces and to accommodate flow obstacles such as baffles.

Given the initial and boundary conditions, and the initial configuration of the liquid surface, the program solves the governing equations and calculates the u and v velocities, the pressure field, and the configuration of the liquid surface, all as functions of time. The program includes an automatic time-step adjustment subroutine in order to maintain numerical stability. It is also capable of supporting slight compressibility of the fluid, surface tension effects, and the simultaneous motion and interaction of two immiscible fluids.

SOLA-VOF is ideal for the present application because of its ability to handle transient flows and free liquid surfaces which is the case with the lading in rail tank cars. In addition, its ability to account for internal obstructions facilitates modelling of the internal shell considered in this study.

There are two drawbacks to the use of the code. Firstly, it is a two dimensional code, whereas the fluid motion within the tank is fully three dimensional. Secondly, there is a lack of available experimental data on wall pressures caused by sloshing in tank cars which would be useful in validating the calculated results. These points are further discussed in the subsections which follow.

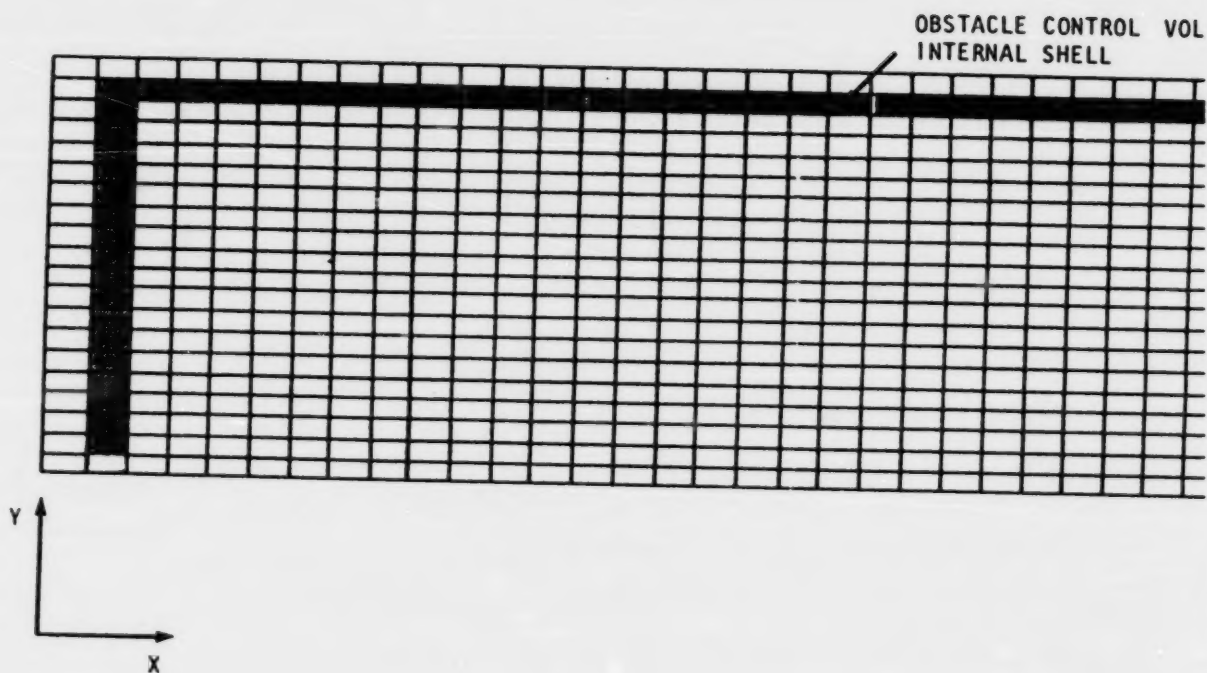
The solution method involved the use of the two dimensional SOLA-VOF code to predict the pressures in the longitudinal vertical plane through the centre of an internal shell equipped tank. The pressures in this plane were then applied (in a conservative manner) to the remaining surfaces within the tank. The two dimensional model applies only at the above mentioned plane because it is a plane of symmetry (no gradients in a transverse sense).

2.3.3 Solution Domain and Assumptions

The exterior shell of the tank car was modelled as a rectangle with an aspect ratio of 6:1 (see Figure 2.3.3-1). The solution domain was divided into a grid of rectangular control volumes: the length (horizontal x-coordinate) of the tank was subdivided into 60 equal sized divisions, and the width (vertical y-coordinate) had 20 equal subdivisions. Hence, each longitudinal subdivision represents 0.305 m (1 ft), and each vertical subdivision represents 0.152 m (6 in.). The domain was thus a model of the longitudinal, vertical centerline of the tank. The internal shell was modelled by flagging certain control volumes as obstacles. The inner shell was modelled so that there was one control volume between the shell and the outer wall at the ends and the top, and no obstacles for the bottom of the shell (see Figure 2.3.3-1).

It was assumed that the fluid within the tank was incompressible (since the lading considered is a liquid) and surface tension was disregarded. It was assumed that the space above the liquid was empty, i.e., vapour was not considered. The liquid is also considered to be inviscid because of the large solution domain in which viscous effects (boundary and shear layers) would be negligible, even on the scale of one control volume. As a consequence of this, the boundaries and the inner shell were considered to be "free slip" boundaries, i.e., there were no velocity gradients near walls. Gravity acts downward at 1 g (9.81 m/s^2).

By specifying an initial fill level, the program automatically set the fluid configuration as horizontal. At the beginning of the time domain, an initial horizontal velocity of 6.3 m/s (14 mph) was assigned to the fluid, and the program began calculation of the transient fluid parameters.



VOLUMES:

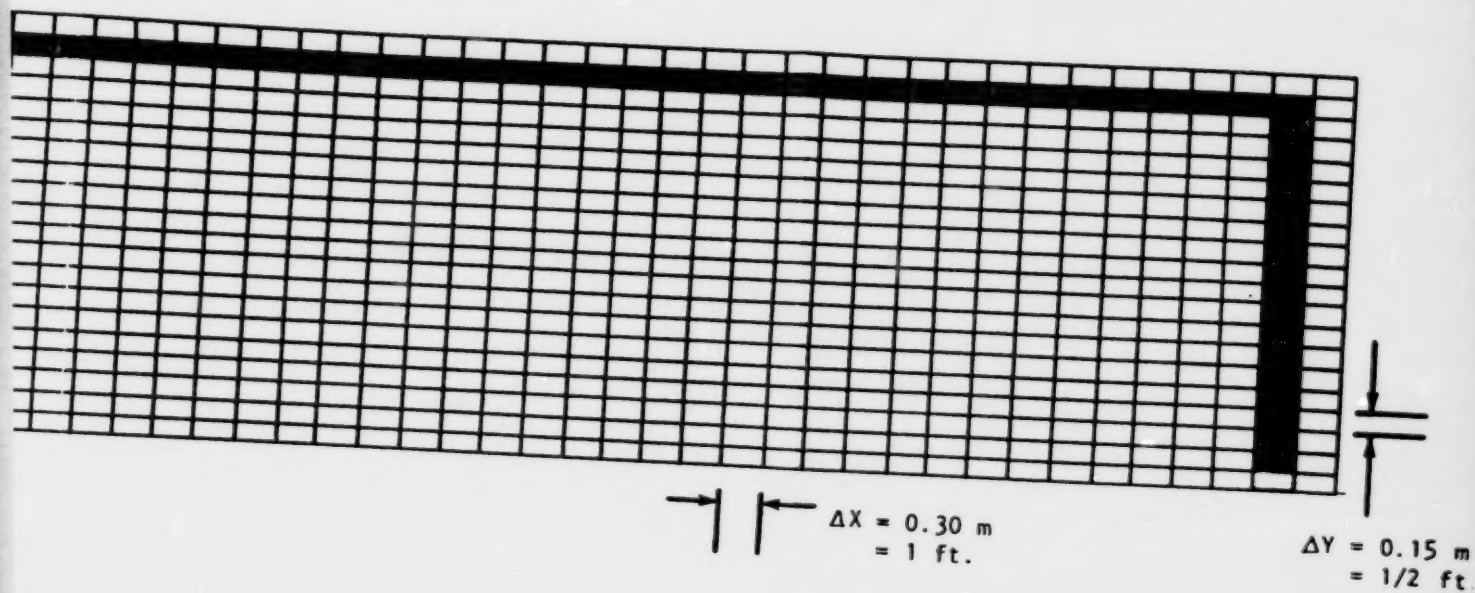


FIGURE 2.3.3-1

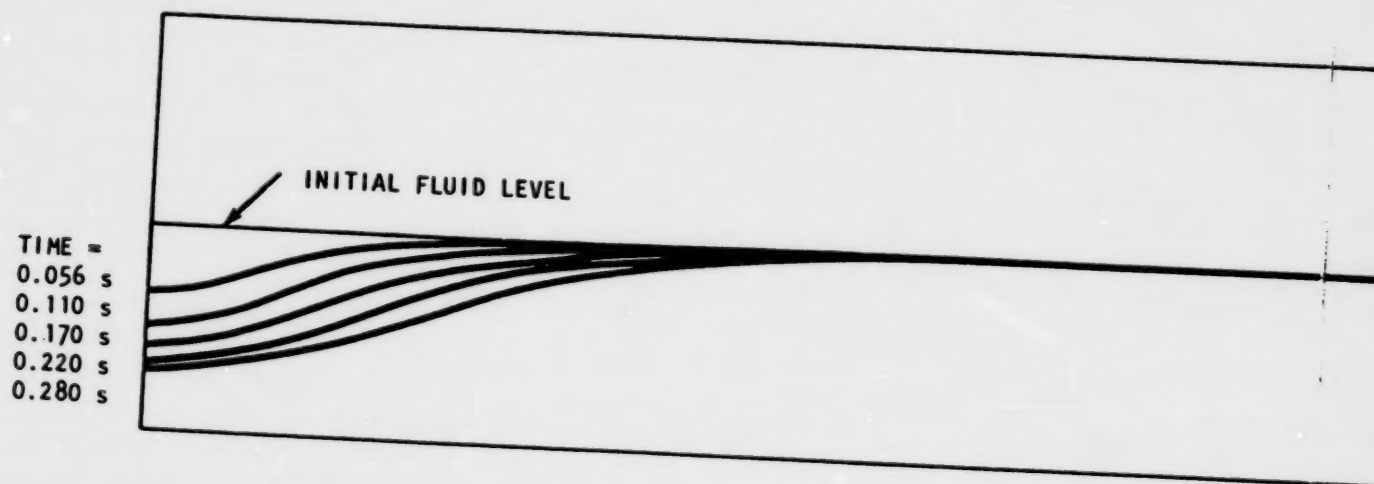
CONTROL VOLUME GRID
FOR SOLUTION DOMAIN

2.3.4 Test Cases

In order to determine whether the results of the SOLA-VOF calculations are reasonable, several test cases were executed. The geometry of the domain and the initial fluid conditions for the test cases were the same as described in the previous subsections with the exception of the removal of the obstacles in the control volumes defining the inner shells. It was assumed that the lading was water (density = 1000 kg/m^3). The initial fluid velocity was 6.3 m/sec (14 mph). Two fill levels were considered, 1.52 m (5 ft.) and 2.74 m (9 ft.) - this corresponds to 50% and 5% outage of a cylindrical tank, respectively.

The results of the calculations are shown in Figures 2.3.4-1 to 2.3.4-2 for both initial fill cases. Figures 2.3.4-1 and 2.3.4-2 show the configuration of the fluid surface at times of 0.056 s , 0.110 s , 0.170 s , 0.220 s and 0.280 s . Illustrations of the general fluid motion within the tank at time = 0.280 s are displayed in Figures 2.3.4-3 and 2.3.4-4. The arrows in 2.3.4-3 and 2.3.4-4 do not indicate relative velocities. The figures are intended to show how SOLA-VOF simulates the evolution of the fluid surface and how it moves within the tank.

The pressure distribution on the walls of the tank are shown in Tables 2.3.4-1 and 2.3.4-2 at times of 0.42 s and 0.11 s for the head and top surface of the tank, respectively. These were the times at which the pressures on the head of the tank were maximum. The keys to the control volume labelling are given in Figures 2.3.4-5 and 2.3.4-6. The tables show that the pressure distribution is fairly uniform along the head and top surface. The total integrated force on the head of the tank, which provides a measure of the momentum impulse on the tank, is also given in the tables. The forces are 290 kN ($65,000 \text{ lbf}$) and 2600 kN ($590,000 \text{ lbf}$) for the 1.52 m and 2.74 m initial fill cases, respectively.



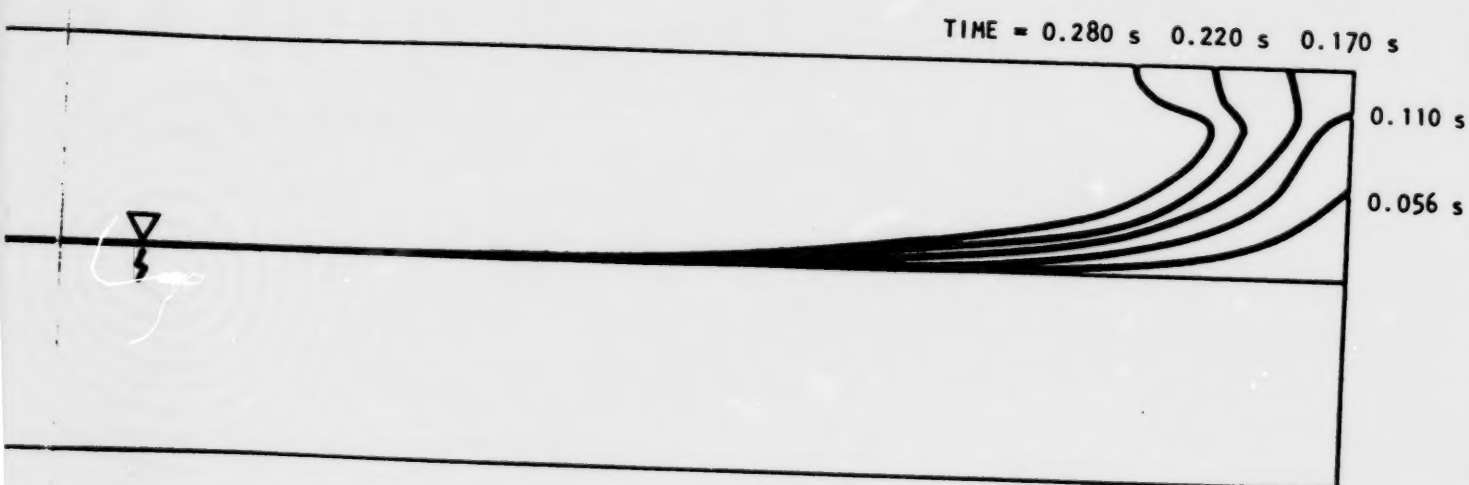
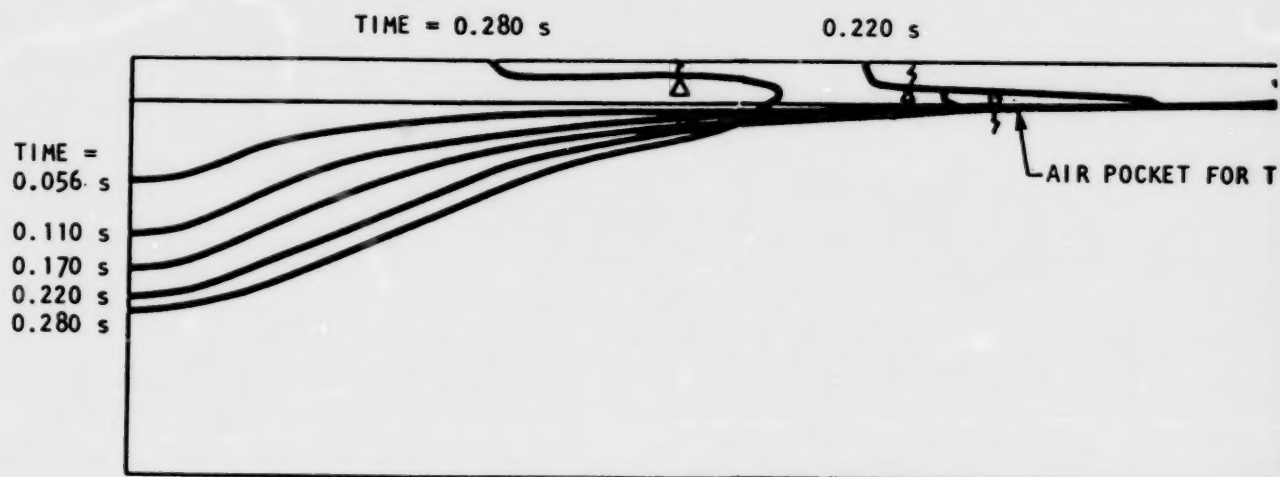


FIGURE 2.3.4-1 TRANSIENT FLUID CONFIGURATION
FOR TANK CAR WITHOUT INTERNAL
SHELL - 1.52 m FILL CASE



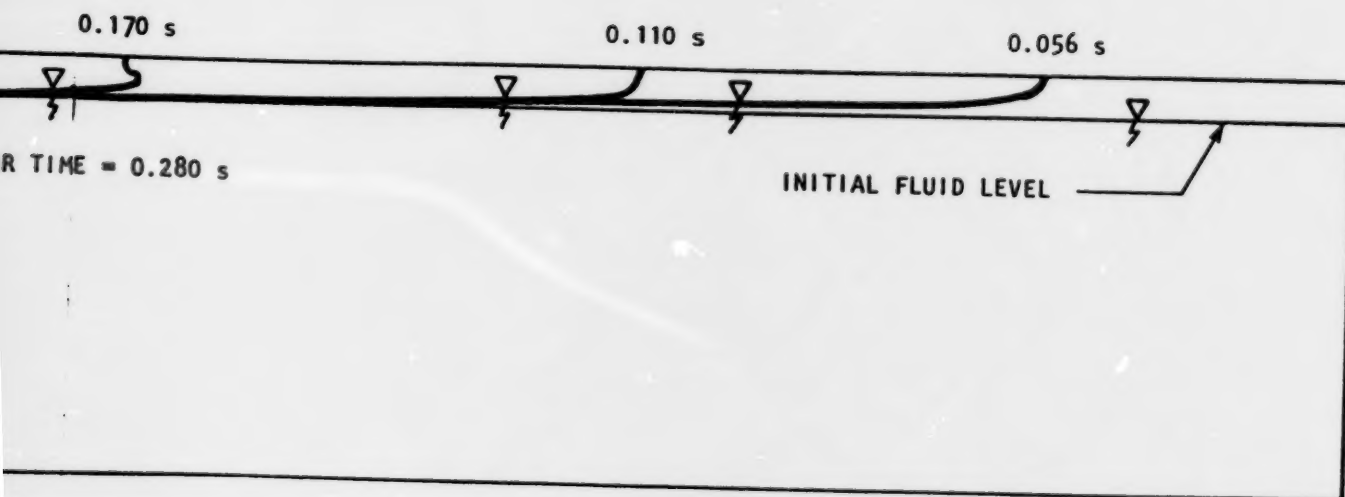
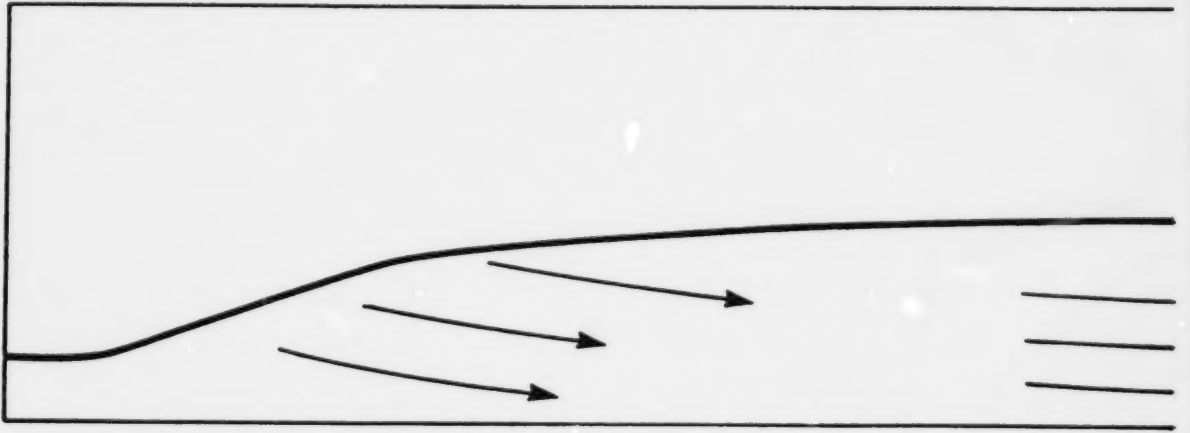
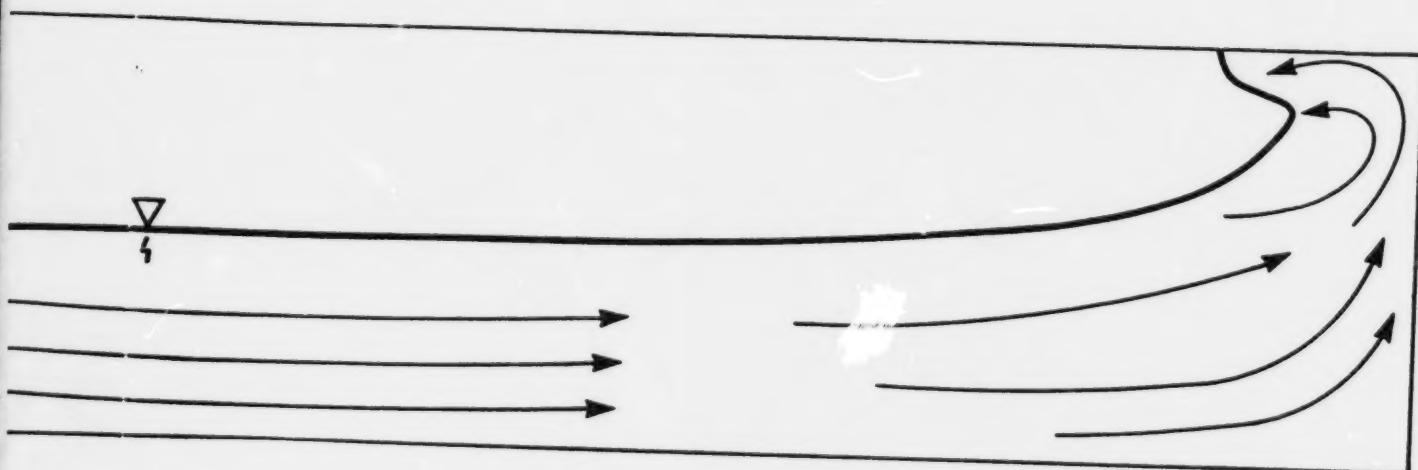


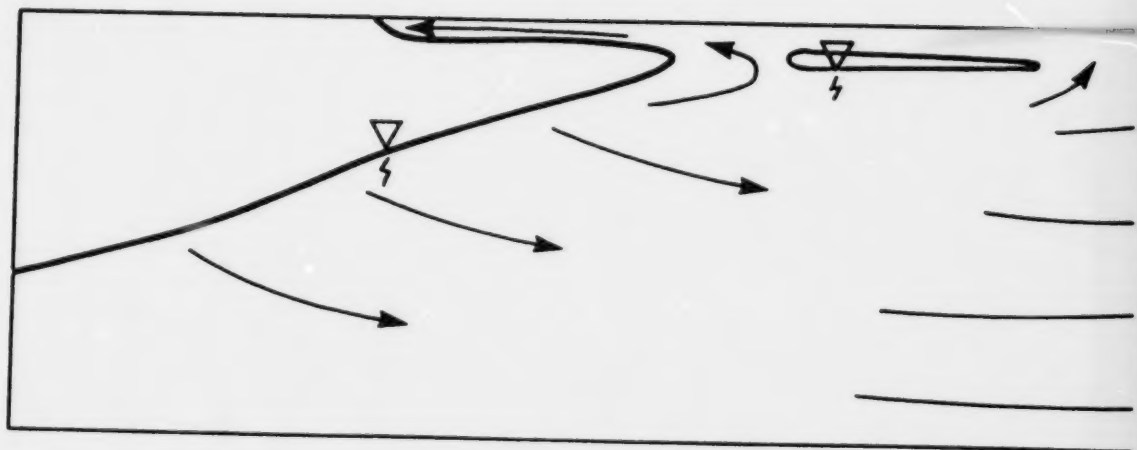
FIGURE 2.3.4-2 TRANSIENT FLUID CONFIGURATION
FOR TANK CAR WITHOUT INTERNAL
SHELL - 2.74 m FILL CASE

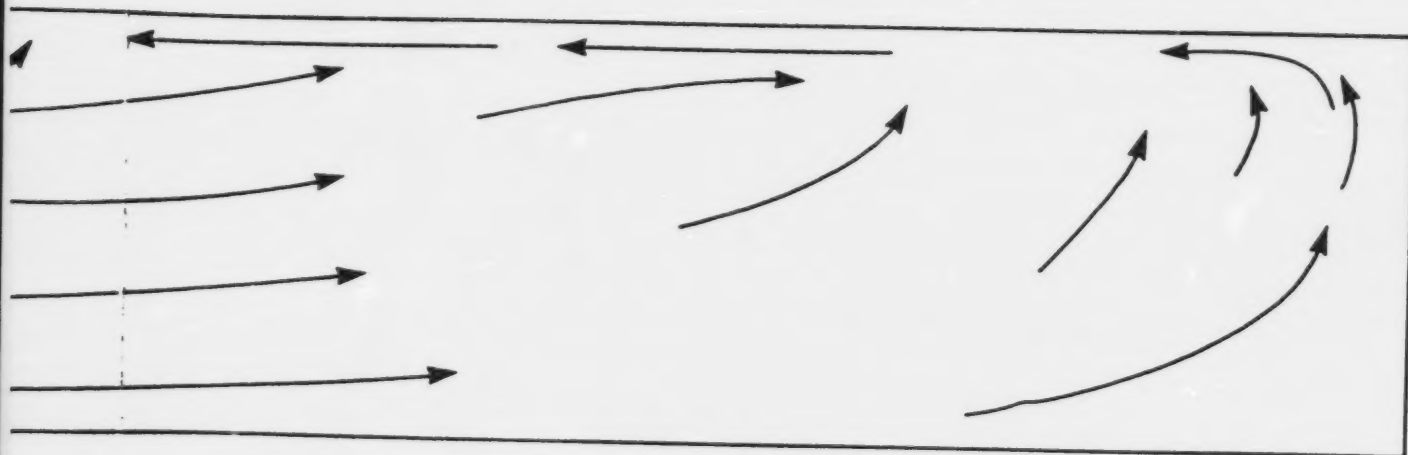




N.B. LENGTH OF ARROWS DOES NOT
INDICATE RELATIVE VELOCITIES

FIGURE 2.3.4-3 FLUID MOTION FOR 1.52 m
FILL CASE - TIME = 0.280 s





N.B. LENGTH OF ARROWS DOES NOT
INDICATE RELATIVE VELOCITIES

FIGURE 2.3.4-4 FLUID MOTION FOR 2.74 m
FILL CASE - TIME = 0.280 s

CONTROL VOLUME**	PRESSURE (kPag)	
	1.52 ⁺ m	2.74 ⁺ m
1	37	350
2	38	350
3	38	350
4	38	350
5	37	350
6	37	350
7	37	350
8	37	360
9	37	360
10	38	360
11	39	360
12	39	360
13	40	360
14	41	360
15	42	360
16	44	360
17	45	360
18	46	360
19	48	360
20	49	370
TOTAL FORCE (kN)	290	2600
TIME (s)	0.42	0.11

** Refer to Figure 2.3.4-5 for the Control Volume Labelling

+ Initial Fluid Level

TABLE 2.3.4-1 WALL PRESSURE DISTRIBUTION AND SLOSH LOAD FOR
TANK CAR WITHOUT INTERNAL SHELL - HEAD

CONTROL VOLUME**	PRESSURE (kPag)	
	1.52 ⁺ m	2.74 ⁺ m
1	37	350
2	37	350
3	36	350
4	33	350
5	24	350
6	11	350
7	1.4	350
8	-1.3	360
9	-1.3	360
10	0.1	360
11	*	360
12	*	370
13	*	380
14	*	400
15	*	420
16	*	460
17	*	470
18	*	350
19	*	-40
20	*	*
TIME (s)	0.42	0.11

** Refer to Figure 2.3.4-6 for the Control Volume Labelling

+ Initial Fluid Level

* No Fluid contained in these Control Volumes

TABLE 2.3.4-2 WALL PRESSURE DISTRIBUTION AND SLOSH LOAD FOR
TANK CAR WITHOUT INTERNAL SHELL - TOP

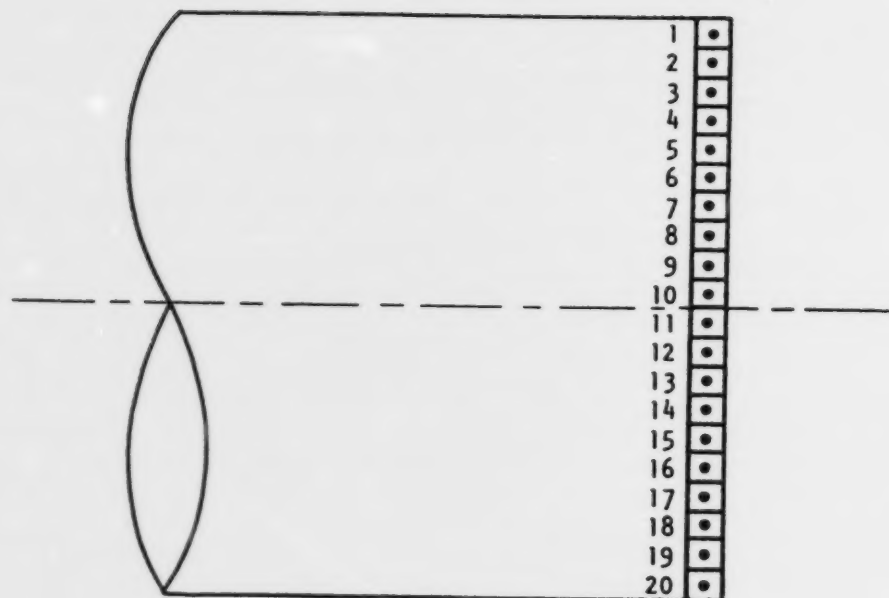


FIGURE 2.3.4-5 KEY TO CONTROL VOLUME LABELLING FOR TANK
WITHOUT INTERNAL SHELL - HEAD

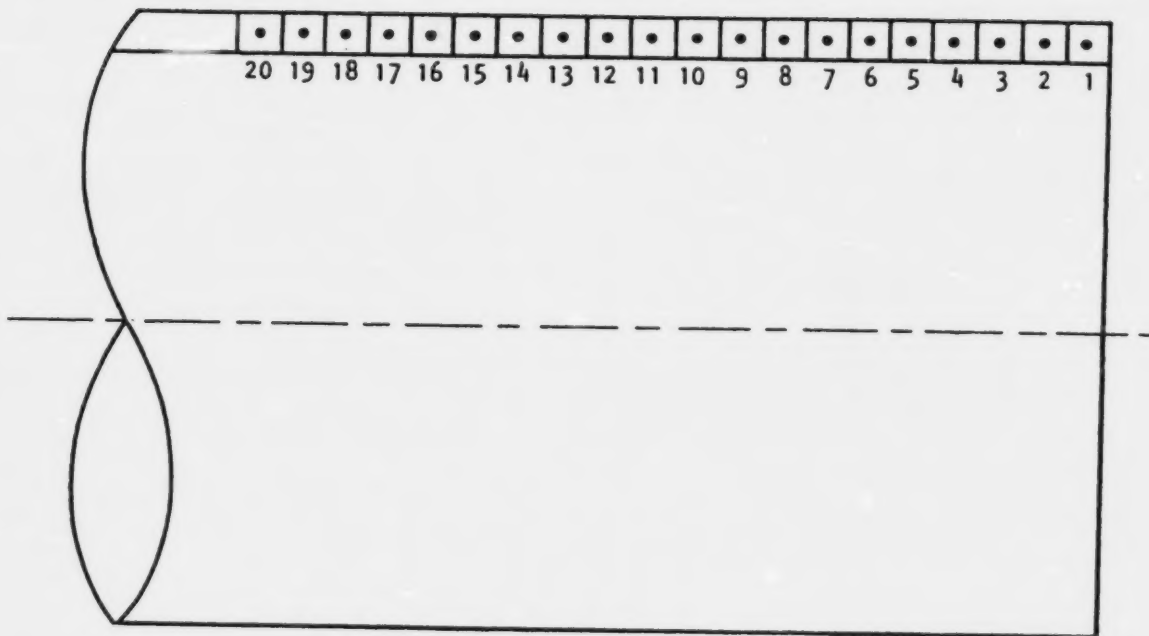


FIGURE 2.3.4-6 KEY TO CONTROL VOLUME LABELLING FOR TANK
WITHOUT INTERNAL SHELL - TOP

2.3.5 Validation of SOLA-VOF Predictions

The technical documentation of SOLA-VOF (2,3) presents several test cases for the code in which the calculations compared favourably with the observed experimental results. In particular, the code was extremely reliable in predicting fluid configurations in complex flows involving barriers. Hence, from the standpoint of prediction of fluid configuration, the code has been well validated.

In order to produce further confidence in the capabilities of SOLA-VOF, it is desirable to compare predictions with actual experimentally observed sloshing loads. There has been no literature in the public domain found which describes experiments measuring wall pressures under slosh conditions in either rail or highway tankers.

An example of typical related work is found in References (4) and (5), in which experimental wall pressure distributions of fluids sloshing in a vertical, cylindrical container under forced oscillations are reported. Qualitatively, those experiments and the calculations presented above indicate similar trends: the experiments show that the greatest slosh pressures occur near the free surface, and that the bottom portion of the fluid acts as a rigid body (uniform pressure distribution). This trend is evident in the predicted pressure distributions shown in Table 2.3.4-2 for the 2.74 m initial fill case. Quantitative comparisons, however, could not be made in this case due to the oscillatory nature of the motion of the experiments, and the geometry of the experimental model.

2.3.6 Numerical Simulation of Tank With Inner Shell

Three initial fill cases were simulated for the system representation described in subsection 2.3.3: 1.52 m (5 ft.), 2.44 m (8 ft.), and 2.74 m (9 ft.), which correspond to 50%, 15% and 5% outage cases, respectively. The

loading was assumed to be propane (density = 500 kg/m^3), and the initial fluid velocity was 6.3 m/s (14 mph).

The calculations showed that the maximum pressures on the head of the inner shell occurred at approximately 0.280 s , 0.220 s and 0.028 s for the 1.52 m , 2.44 m and 2.74 m initial fill cases, respectively. Figure 2.3.6-1 displays the fluid configurations at those times, and the wall pressures on the head and top surfaces are listed in Tables 2.3.6-1 and 2.3.6-2, respectively. The key to the control volume labelling is shown in Figures 2.3.6-2 and 2.3.6-3.

2.3.7 Interpretation of Results

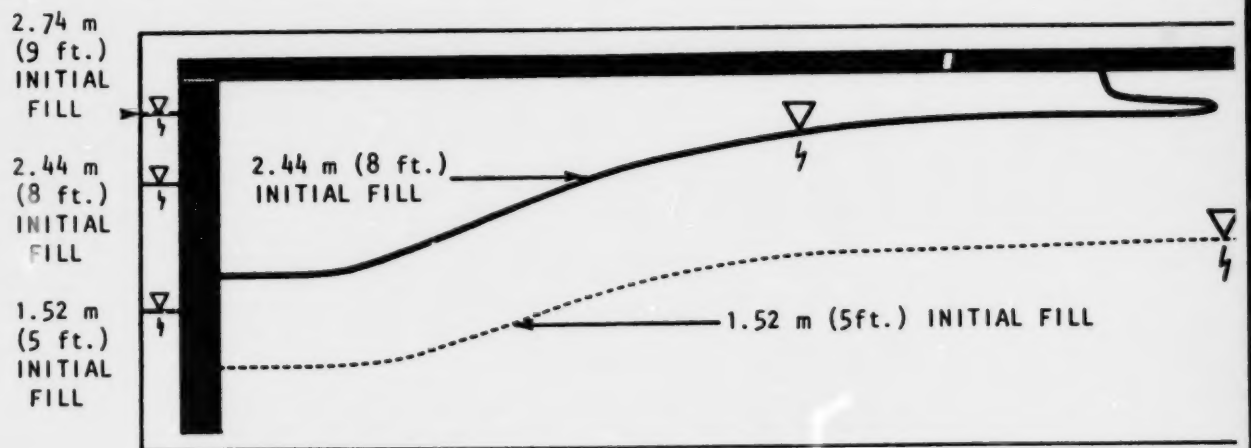
The results introduced in the previous section show that for a tank without the internal shell and filled to 95% full with water the overpressures range from $40\text{--}470 \text{ kPa}$ ($5\text{--}70 \text{ psi}$). For the tank with internal protection, filled to 95% full with LPG the overpressures range from $20\text{--}170 \text{ kPa}$ ($3\text{--}25 \text{ psi}$).

LPG is half as dense as water and therefore to compare the pressures for the internally protected tank with those of the unprotected tank it is necessary to double the pressures for the former (i.e., $40\text{--}340 \text{ kPa}$). The range of pressures for the two cases should be similar because the presence of the internal shell will not effect the local overpressures in the inner cavity significantly. This is apparent from the results presented. However, this is not to say that the slosh loads on the internal shell are the same as those experienced by the primary shell in an unprotected tank. The net pressure applied to the internal shell is equal to the difference between the pressures applied to the inner and outer surfaces of the inner shell, and as indicated in the results, the pressures in the annulus offset the pressures in the inner cavity.

As stated earlier, the SOLA VOF predictions only apply on the longitudinal vertical plane through the tank centre-line. It was not possible, using the existing version of SOLA VOF, to predict the full three dimensional flow in the annulus and inner cavity. However, the structural analysis requires the slosh pressures throughout the tank.

INITIAL F
LEVEL

1.52 m
2.44 m
2.74 m



ELAPSED
TIME

0.28 s
0.22 s
0.03 s

2.44 m
(8 ft.)
INITIAL
FILL

2.74 m
(9 ft.)
INITIAL
FILL
1.52 m
(5 ft.)
INITIAL
FILL

N.B. THERE IS NO FREE SURFACE
INSIDE INTERNAL SHELL FOR
2.74 m (9 ft.) INITIAL FILL

FIGURE 2.3.6-1 FLUID CONFIGURATION FOR TANK
CAR WITH INTERNAL SHELL

CONTROL VOLUME **	PRESSURE (kPag)			CONTROL VOLUME **	PRESSURE (kPag)		
	1.52 ⁺ m	2.44 ⁺ m	2.74 ⁺ m		1.52 ⁺ m	2.44 ⁺ m	2.74 ⁺ m
1	25	130	97	A	*	62	140
2	24	130	98	B	*	63	140
3	23	130	99	C	*	64	130
4	22	130	100	D	*	65	120
5	21	130	100	E	*	66	120
6	20	130	100	F	0	67	110
7	20	130	100	G	0.7	68	100
8	20	130	100	H	1.4	68	97
9	20	140	110	I	2.2	69	90
10	20	140	110	J	2.9	70	84
11	20	140	110	K	3.6	71	77
12	21	140	110	L	4.3	72	71
13	21	140	110	M	5.0	73	64
14	22	140	110	N	5.7	74	57
15	22	140	120	O	6.5	75	51
16	23	140	120	P	7.2	75	44
17	23	140	120	Q	7.9	76	38
18	23	140	120	R	8.6	77	31
				S	9.3	78	25
				T	14.0	94	61
TIMES (s)	0.28	0.22	0.03	TIME (s)	0.28	0.22	0.03

** Refer to Figure 2.3.6-2 for the Control Volume Labelling

+ Initial Fluid Levels

* No Fluid contained in these Control Volumes

TABLE 2.3.6-1 WALL PRESSURE DISTRIBUTION FOR TANK CAR
WITH INTERNAL SHELL - HEAD

CONTROL VOLUME **	PRESSURE (kPag)			CONTROL VOLUME **	PRESSURE (kPag)		
	1.52 ⁺ m	2.44 ⁺ m	2.74 ⁺ m		1.52 ⁺ m	2.44 ⁺ m	2.74 ⁺ m
1	25	130	97	29	*	140	58
2	24	130	97	30	*	150	57
3	18	130	97	31	*	150	55
4	7.5	130	96	32	*	160	53
5	7.0	130	96	33	*	170	52
6	*	130	94	34	*	140	50
7	*	130	92	35	*	*	49
8	*	130	91	36	*	*	47
9	*	130	90	37	*	*	46
10	*	130	88	38	*	*	44
11	*	130	87	39	*	*	43
12	*	130	85	40	*	*	41
13	*	130	83	41	*	*	39
14	*	130	82	42	*	*	38
15	*	130	80	43	*	*	36
16	*	130	79	44	*	*	35
17	*	130	77	45	*	*	33
18	*	130	76	46	*	*	32
19	*	130	74	47	*	*	30
20	*	130	72	48	*	*	29
21	*	130	71	49	*	*	28
22	*	130	69	50	*	*	26
23	*	130	68	51	*	*	25
24	*	130	66	52	*	*	24
25	*	130	64	53	*	*	23
26	*	130	63	54	*	*	22
27	*	130	61	55	*	*	21
28	*	130	60	56	*	*	21
A	*	62	140	D	*	*	*
B	*	0.7	*	E	*	*	*
C	*	0.3	*	F	*	*	*
TIMES (s)	0.28	0.22	0.03	TIME (s)	0.28	0.22	0.03

** Refer to Figure 2.3.6-3 for the Control Volume Labelling

+ Initial Fluid Levels

* No Fluid contained in these Control Volumes

TABLE 2.3.6-2 WALL PRESSURE DISTRIBUTION FOR TANK CAR WITH
INTERNAL SHELL - TOP

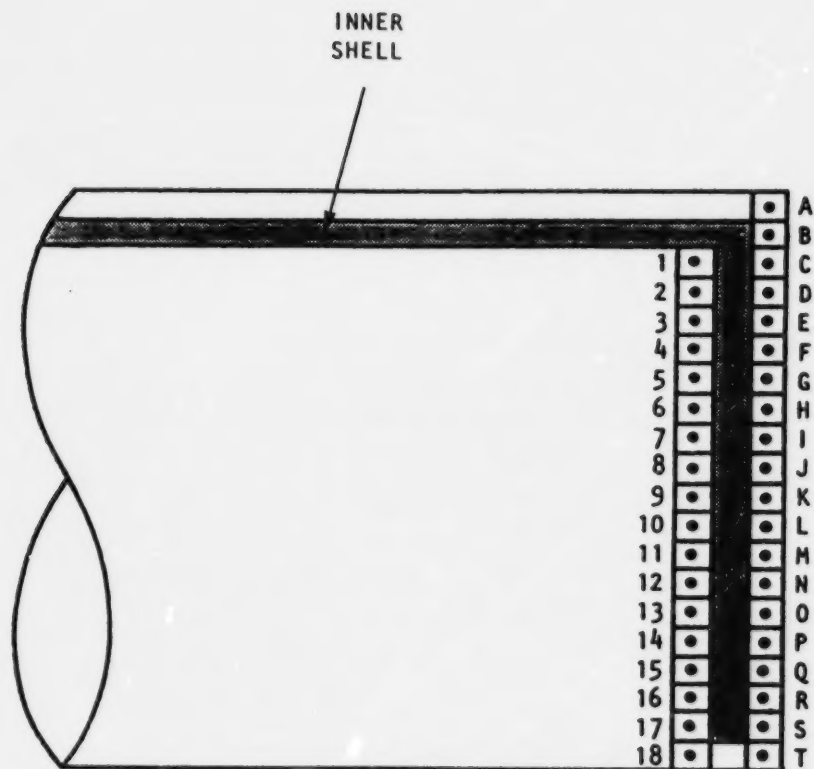


FIGURE 2.3.6-2 KEY TO CONTROL VOLUME LABELLING FOR TANK CAR WITH INTERNAL SHELL - HEAD

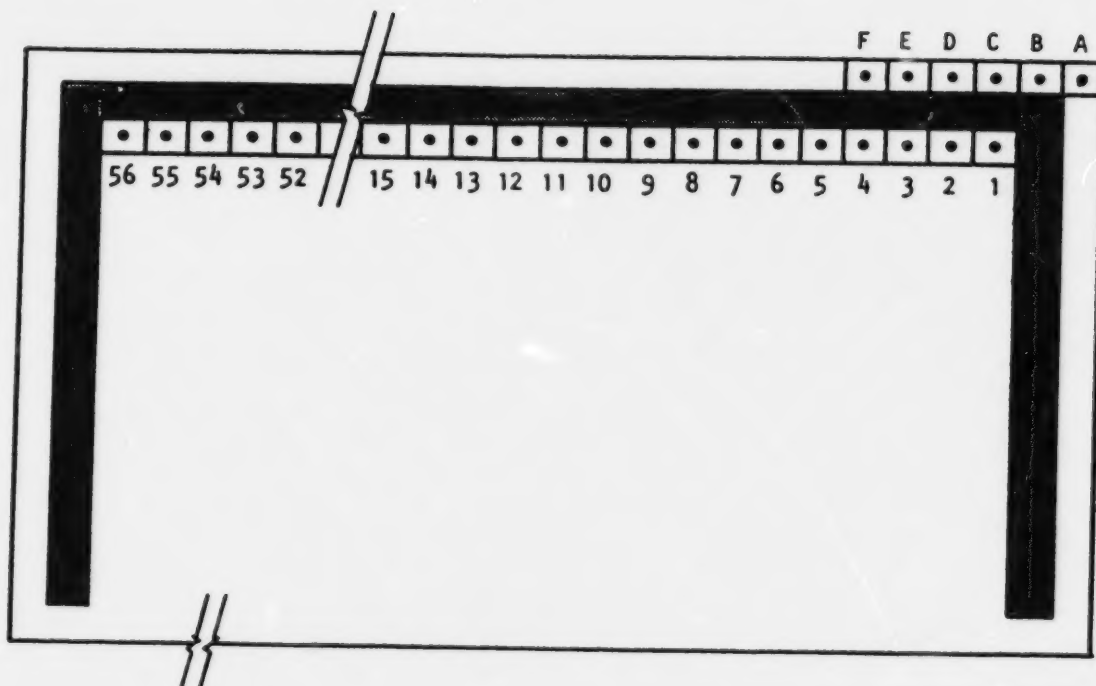


FIGURE 2.3.6-3 KEY TO CONTROL VOLUME LABELLING FOR TANK CAR WITH INTERNAL SHELL - TOP

Therefore, it was necessary to approximate the three dimensional pressure distribution using the two dimensional SOLA VOF results.

For the end cap of the internal shell, it was assumed that the pressure on the end cap centre-line for a given height applied equally to all other points on a line across the width of the end. This is illustrated in Figure 2.3.7-1.

For all other longitudinal positions the pressures calculated at the top of the calculation plane were applied around the entire circumference of the inner shell. This procedure is only applied to longitudinal positions where the inner surface of the inner shell at the top is wetted by liquid. Other longitudinal positions do not experience slosh loads because they are not in contact with liquid. In the annulus the pressures are assumed to be zero for all circumferential positions even if wetted by liquid. Figure 2.3.7-2 illustrates the above methodology.

The overall structural analysis will be conservative when using the SOLA-VOF predictions in this manner. The reasons for this are as follows:

- 1) The two dimensional SOLA VOF pressure predictions will be conservative because the two dimensional model does not account for fluid initially in the annulus. It only accounts for fluid moving from the inner cavity to the annulus. Therefore the offsetting effect of the annular pressures are underpredicted.
- 2) The application of the pressure at the top of the inner shell to all circumferential locations at a fixed longitudinal position is conservative because it ignores the fact that the pressure in the internal shell must drop as you move around the shell to the bottom opening (i.e., the pressure on the inner surface and outer surface of the shell must be equal at the bottom because of the presence of the gap at the bottom of the internal shell). Therefore, the total force applied to the internal shell is overpredicted.
- 3) The assumption that the pressure is zero in the annulus along the length of the tank is conservative because it neglects the pressure offset caused by liquid in the annulus. This again results in an overpredicting of the loads on the inner shell.

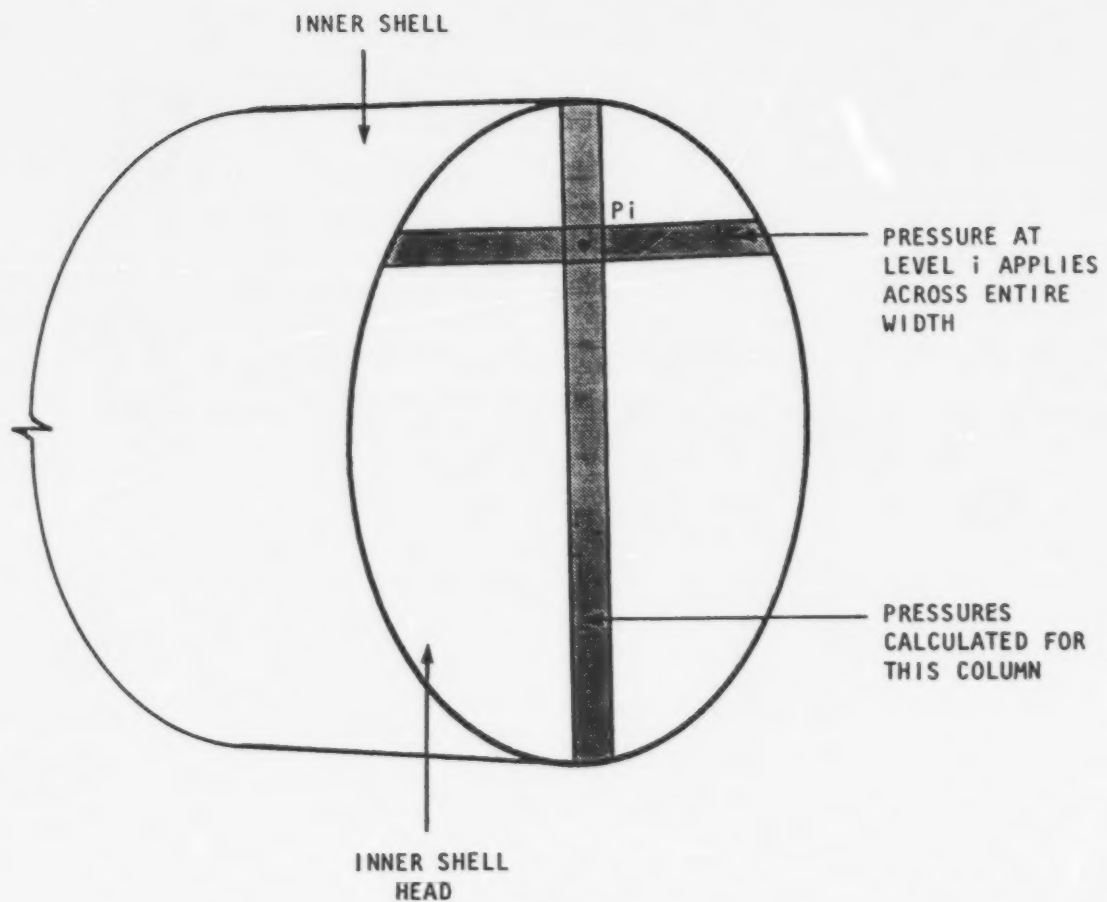
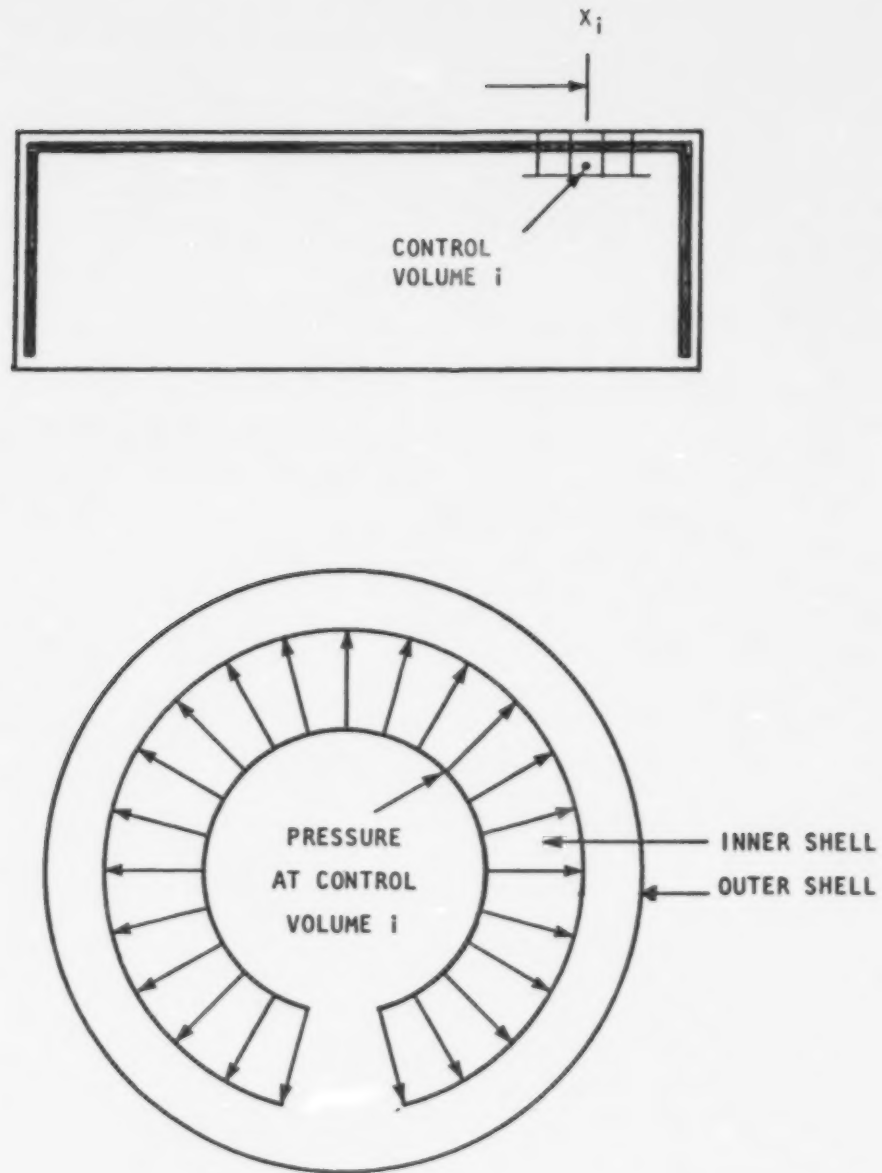


FIGURE 2.3.7-1 AREA OF EFFECTIVE PRESSURE APPLICATION FOR INTERNAL SHELL HEAD



CROSS-SECTION AT CONTROL VOLUME i

FIGURE 2.3.7-2 CIRCUMFERENTIAL PRESSURE DISTRIBUTION FOR
INTERNAL SHELL

2.4 ANALYSIS RESULTS

A variety of load cases for both empty and laden tank cars have been analyzed. This section presents the results of these analyses.

2.4.1 Maximum Stress

Table 2.4.1-1 shows the stresses in the inner shell obtained from the finite element analysis. For each load case analysed, two stresses are given. One is the maximum stress in the shell itself. The other value is the maximum stress in the spacer pins which separate the inner and outer shells. The inner shell stresses given are based on Von Mises-Hencky theory. This theory is widely accepted and is the most common stress theory for predicting yield in ductile materials. For the spacer pins the stress quoted is the maximum normal stress.

For TC-128B steel used in tank cars, the yield strength is from 345 MPa to 450 MPa depending on the heat treatment used. The maximum stress of 159 MPa occurs for the 2.44 m initial fluid level sloshing case. This stress is well below yield but because this load will occur relatively frequently, fatigue must be considered. Appendix A contains a fatigue calculation for the shell/spacer interface. This calculation shows that the inner shell is capable of withstanding 90000 cycles of the worst case loading. This is equivalent to six cycles a day for forty years, which should be more than sufficient. The pressure distribution assumed for the sloshing loads was much more severe than would be expected. It is probable that a three dimensional sloshing analysis would produce lower stress values which would greatly increase the fatigue life. For the sloshing load cases, the peak stresses occur at the support points along the edge of the inner shell slot.

2.4.2 Shell Thickness/Support Spacing

The shell thickness and support layout are entirely governed by the loads caused by fluid sloshing. From the analysis performed, a shell thickness of 3.2 mm (1/8 in.) is required. The required support layout is shown in Figure 2.4.2-1. Supports are required every 1.22 m (4 ft.) axially and radially every 30°. Along the slot, supports are required every 0.30 m (1 ft.). In addition,

LOAD CASE	FILL CONDITION	LOCATION	MAXIMUM STRESS (MPa)
Sloshing	Laden (2.74 m*)	Inner Shell	101
		Spacers	92
	Laden (2.44 m*)	Inner Shell	159
		Spacers	143
	Laden (1.52 m*)	Inner Shell	14.5
		Spacers	26.2
Inverted Tank	Laden	Inner Shell	15.2
		Spacers	30.3
Axial (12 g)	Empty	Inner Shell	5.1
		Spacers	16.6
Vertical (3 g)	Empty	Inner Shell	1.7
		Spacers	4.1
Lateral (2 g)	Empty	Inner Shell	1.6
		Spacers	2.4

* Initial Fluid Level

TABLE 2.4.1-1 MAXIMUM STRESS IN INNER SHELL AND SPACER PINS

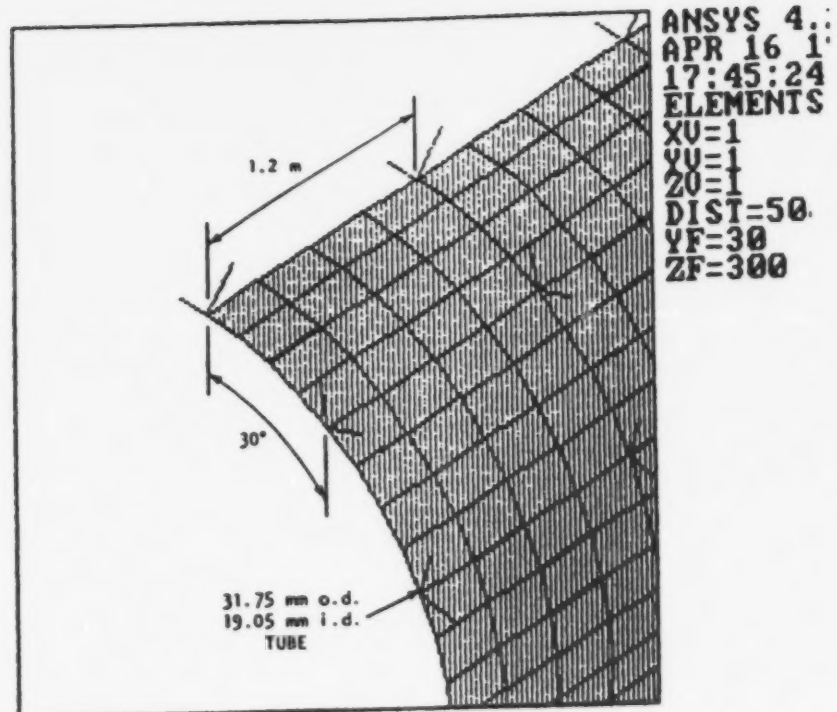


FIGURE 2.4.2-1 GENERAL SUPPORT LAYOUT

a horizontal support is required every 0.61 m (2 ft.) across slot on the bottom of the inner shell between supports. Finally, it will be necessary to have a member along the edge of the slot in the axial direction (both sides). Figure 2.4.2-2 shows the layout of support members along the slot. For the end cap, spacers are required at the locations shown in Figure 2.4.3-3. All members along the slot (Figure 2.4.2-2) are solid 38 mm (1½ in.) diameter bars. Everywhere else (including between end caps) 32 mm (1¼ in.) outside diameter, with 6.4 mm (¼ in.) wall thickness tubing will be used.

As mentioned previously, a three dimensional sloshing analysis will likely show a pressure of zero along the slot. This will reduce the support requirements along the slot considerably.

The total weight added to the tank car by the inner shell and the support scheme will be 47.1 kN (10,600 lb). This will increase the weight of the empty tank car by 10%. By comparison, the jacket and insulation (without the head shields) weighs 46.7kN (10,500 lb) (20). The increase in weight will not affect the tank cars carrying capacity for the lighter fluids like propane since the tank car will be below the rail load limit when entirely filled. It will lower the tank's capacity for more dense fluids such as vinyl chloride.

2.4.3 Natural Frequencies

A variety of forcing functions are possible during normal operation of a tank car. Table 2.4.3-1 shows some examples of force inputs which are likely to be significant. Frequencies were calculated assuming a three foot diameter wheel, and the wheel spacing for a CTC-112J340W tank car.

Table 2.4.3-2 presents the first five natural frequencies for the inner shell and support scheme as discussed in this section. The lowest natural frequency of the inner shell is higher than the forcing function frequencies presented in Table 2.4.3-1, therefore resonance of the inner shell should not be a problem.

The frequencies presented in Table 2.4.3-2 are for an empty tank car. For a laden car, the frequencies would change, but presumably the fluid would damp out any induced vibrations.

A much more complex situation is the dynamic interaction of the primary and inner shells. No measured vibration spectra could be obtained from an over-the-road environment. There are certain general comments which can be drawn however.

The tank car will have a natural frequency on its suspension system in the 1-5 hz range, depending upon the type of suspension and weight of lading. Any impact to the tank in this frequency range will not be a problem since the lowest natural frequency of the tank shell structures is above this range. If low frequency oscillation of the primary tank on its suspension is currently not a problem, one can infer that there will be even less of a problem with the inner tank in place.

The suspension system also attenuates the level of any vibration at a higher frequency than its natural frequency. As such, high frequency vibration is likely to be of relatively low amplitude.

If the high frequency vibration is transmitted directly through the primary shell to the inner shell, no problems are foreseen in the inner shell. This can be stated since there are currently no problems in the primary shell with resonances due to high frequency vibration and the same vibration amplification factor at resonance would apply for the inner and outer shells.

There are no problems foreseen in submerged areas of the tank structure due to the high levels of viscous dampening provided by the lading.

There is one possible source of concern involving the dynamic interaction between the inner shell and primary shell. If there were a high frequency mode of vibration excited of the primary shell and this vibration in turn excited a resonance in the inner shell, the amplitude of the input vibration is amplified twice. This could only occur if the inner shell structure happened to have a mode of vibration of the same frequency, location and excitation direction as the primary shell. Although the likelihood of this being the case is remote, it is possible and bears consideration since fatigue problems could be encountered.

A further more detailed analysis of the entire structure will determine whether such a condition would exist with the current design. Actual field testing of a prototype would also reveal this type of problem.

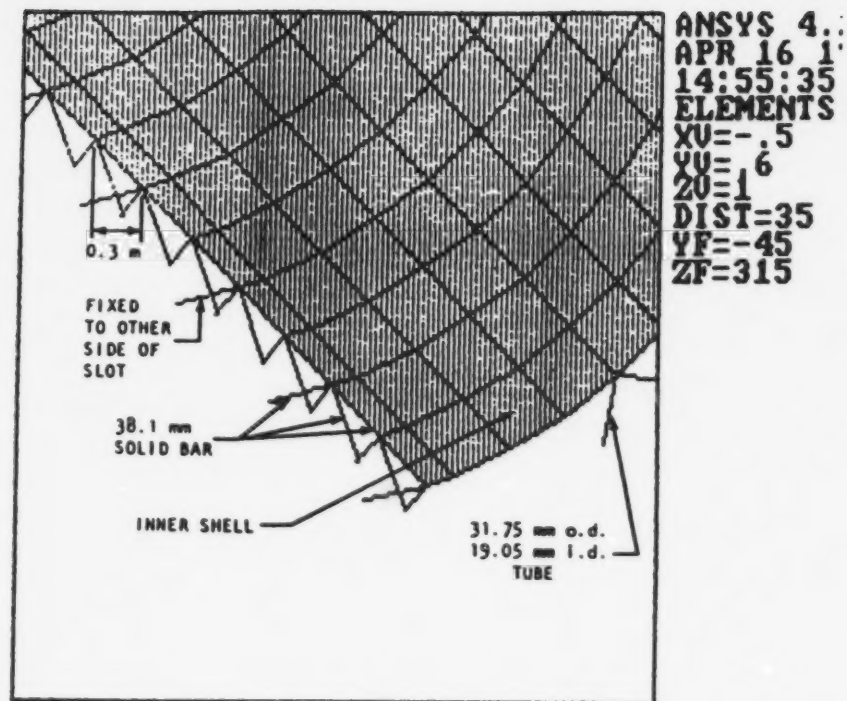


FIGURE 2.4.2-2 SUPPORT LAYOUT AT INNER SHELL GAP

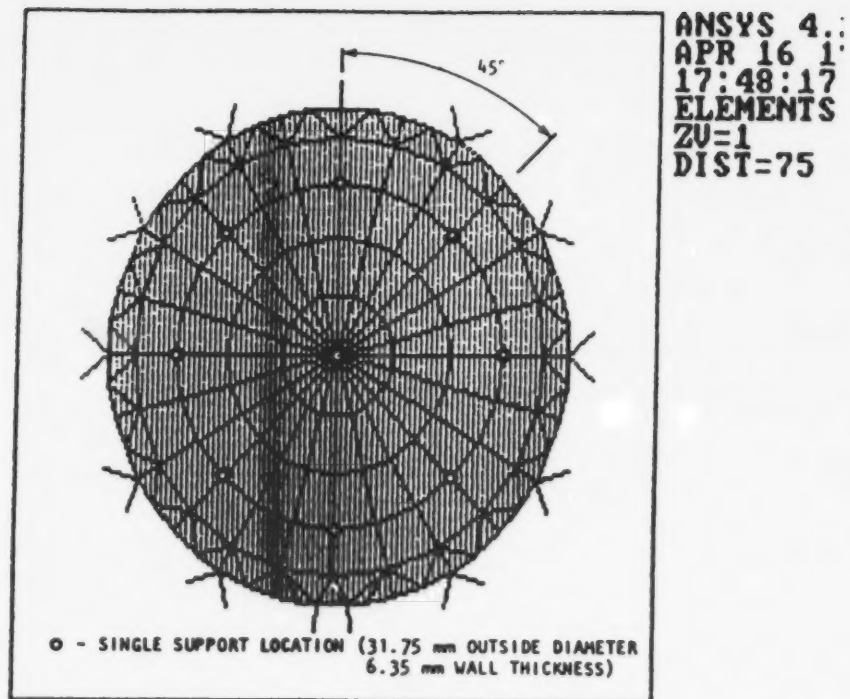


FIGURE 2.4.2-3 END CAP SPACER LOCATIONS

Forcing Function	Train Speed (mph)	Frequency (Hz)
Single Wheel Flat	20	3.1
	40	6.2
	60	9.3
Double Wheel Flat	20	6.2
	40	12.4
	60	18.6
Rail Connection	20	5 max
	40	10 max
	60	15 max

TABLE 2.4.3-1 POSSIBLE FORCING FUNCTIONS AND
CORRESPONDING FREQUENCIES

MODE	FREQUENCY (Hz)
1st	22.6
2nd	24.4
3rd	30.3
4th	40.5
5th	54.8

TABLE 2.4.3-2 NATURAL FREQUENCIES OF INNER SHELL FOR
EMPTY TANK CAR

3.0 OPERATIONAL CONSIDERATIONS

The introduction of an Internal Thermal Protection Device (ITPD) into pressurized rail cars, will effect tank car operations to some extent. This section will address all aspects of these effects, identify problem areas, and place an emphasis upon proposed solutions.

Although the ITPD may be of use in a number of lading applications, this is beyond the scope of the report. All specifications referred to, and alternatives discussed, pertain to CTC-112J Class Tank Cars, carrying LPG or anhydrous ammonia.

The various operational procedures can be classified into four categories:

- ° Daily operations
- ° Semi-yearly
- ° Every ten years
- ° As required

The most significant procedures for each of the categories are summarized in Table 3.0-1. Each of these procedures will be described in more detail in the following sections.

3.1 DAILY OPERATIONS

3.1.1 Tank Car Loading and Unloading

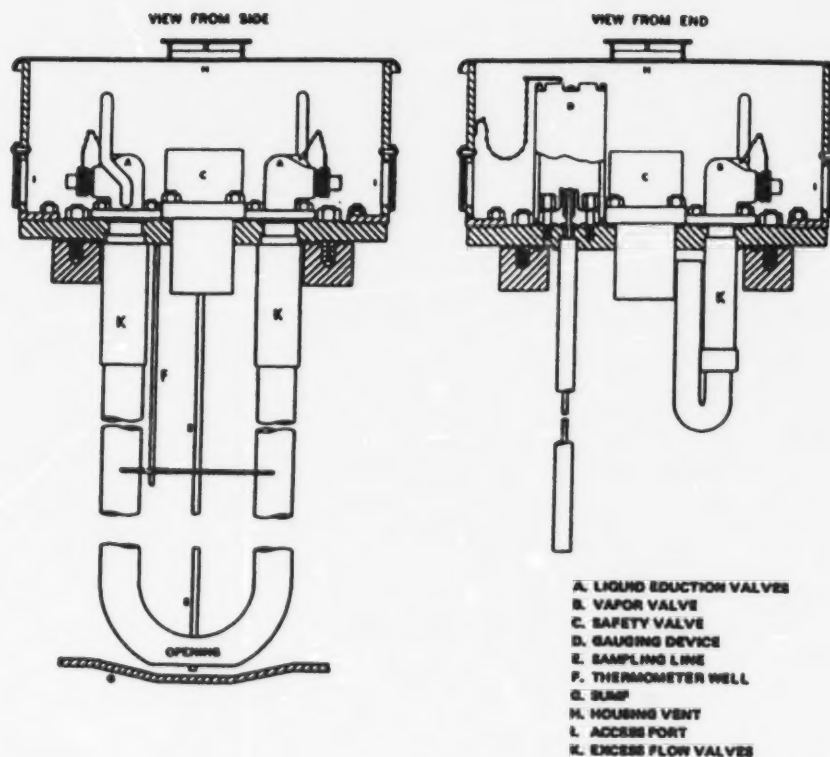
The loading and unloading of LPG and anhydrous ammonia from CTC-112J class tank cars is accomplished through the use of two liquid eduction valves and pipes located on the top part of the car. The equipment needed to accomplish this is housed in the manway in the form of a valve train and fitting arrangement extending into the sump (Figure 3.1.1-1).

Lading handling procedures vary slightly with commodity, but in general the liquid is added or removed from the sump through eduction pipes while the car is kept at the same pressure through the use of venting. The car uses a vapor valve to serve as an equalizing connection between the reservoir and tank ensuring a closed system.

FREQUENCY	PROCEDURE
Daily Operations	- Pre-loading inspection - Loading procedure - Pre-transit inspection - In-transit endurance to applied loads - Pre-unloading inspection - Unloading procedure
Semi-Yearly	- Cleaning tank of old lading - Preparation of tank for new commodity
Every Ten Years	- Corrosion inspection (inside and outside) - Complete valve train inspection - Hydrostatic leak/pressure test
As Required	- Impact damage to exterior shell - Corrosion damage (pitting, blisters) - Postweld heat treatment following - Any welding to the pressure vessel

TABLE 3.0-1 SUMMARY OF OPERATIONAL PROCEDURES

PRESSURE FITTINGS



The illustrations above depict a housing and fittings arrangement for a CTC 112J340 tank car used for such ladings as LPG and anhydrous ammonia. Generally, pressure cars have no bottom outlet so loading and unloading is accomplished through top fittings.

Each car is conventionally loaded and unloaded through two liquid eduction valves and an eduction pipe which reaches to the sump. Also, a vapor valve serves as an equalizing connection between the tank car and the vapor recovery system to facilitate closed system loading and unloading.

The fittings area also includes a spring loaded safety valve, a gauging device used to determine outage, a thermometer well, a sampling line, and various excess flow and check valves which are integral to the fittings.

FIGURE 3.1.1-1 PRESSURE FITTING ARRANGEMENT (REFERENCE 10)

The introduction of an ITPD will create two operational problems:

- (a) valve train location interference, and
- (b) lack of inner shell venting.

The valve train has components that extend to various depths in the tank, with the eductor pipes reaching all the way into the sump. To avoid interference with the inner shell, some sort of passage for the eductors, thermowell and level indicator must be provided. It must be noted that the holes through the inner shell must be sealed properly, or the ITPD would experience a decrease in its ability to direct two-phase flow through the annular gap during a fire situation.

As internal access is also a consideration, this arrangement will be examined in more detail in subsection 3.4.

The problem of adequate venting can be addressed by simply providing a vent passage from the inner shell to the tank shell vent (Figure 3.1.1-2). This vent would probably be a manually operated unit with a handwheel actuator incorporated into the valve train. The procedure would be to open the vent manually before handling lading and closing it to "arm" the ITPD upon completion of the operation. It should be noted that an automatic pressure relief type could be employed, but it could reduce the effectiveness of the unit during a fire.

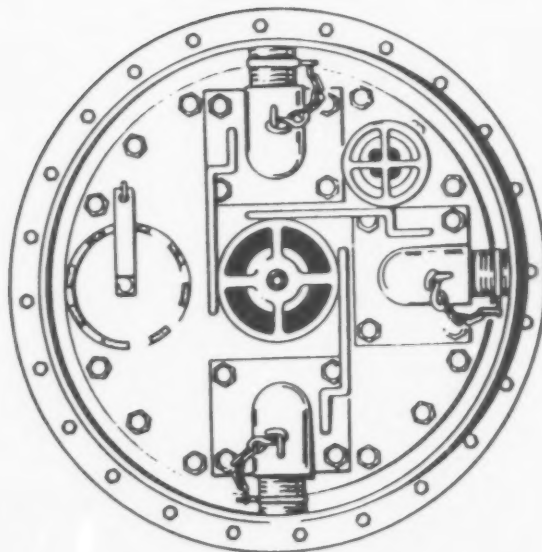
3.1.2 Payload Reduction

Some change is expected in the quantity of lading the cars could transport, due to the addition of the extra weight of the steel used in constructing the device. Two constraints are used to evaluate the payload reduction:

- (a) volume analysis, and
- (b) weight analysis.

Assuming 3 mm (1/8 ") thick plate for the inner shell, and including fixtures, the ITPD would weigh approximately 47 kN (10600 lb). The CTC-112J class tank car is able to carry a full load of commodities such as LPG or anhydrous ammonia which have a low filling density. Therefore, the internal shell would decrease the load only by the volume

TOP VIEW



SIDE VIEW

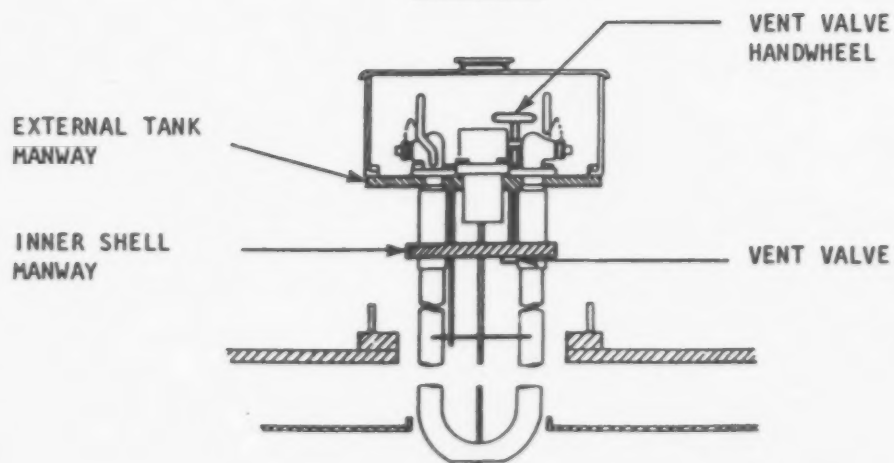


FIGURE 3.1.1-2 INNER SHELL VENT

of lading displaced. In the case of LPG, this is 0.61 m³ (162 US gal), or 0.5% of the 127 m³ (33600 US gal) capacity.

However, if other commodities with higher filling densities, such as vinyl chloride, are to be transported, the normal procedure is to light load the tank car, i.e., not filled to volumetric capacity, so as not to exceed the permissible gross weight of 1.17 MN (263,000 lb) for tank car and lading. Thus, for vinyl chloride, the total payload reduction is 47 kN (10,600 lb) due to the weight of the internal shell. This is 7% of the 690 kN (155,500 lb, Ref. (10)) allowable load in a tank car without internal shell. However, if the jacketting were removed from the CTC-112J (10,500 lb), there would be essentially no payload reduction due to the ITPD. This point should be examined in the event the ITPD is considered for loadings other than those covered in this report.

3.2 IN-TRANSIT OPERATION

From an operational standpoint, the only considerations during transit of a rail tank car fall into three categories:

- (a) walk around and mobile inspection,
- (b) placarded tank car lading identification, and
- (c) stresses induced due to coupling and braking actions.

As operating trains can only be examined by visual inspection, in-transit inspections are basically general checks for equipment damage done by the carrier. More specifically, the CTC Regulations for the Transportation of Dangerous Commodities by Rail requires that loaded tank cars be inspected by the carrier before acceptance at the originating point, and when received at interchanges to see that they are not leaking. Other items examined during a stationary inspection include proper operation of the air and handbrake systems. Journal boxes and trucks are examined for damage as well as walkways, placards, couplers and tank structural arrangement.

Once the train is in motion, station checks are performed along the route. This visual observation looks primarily for dragging items or broken wheels, etc., as the train passes by. The addition of an ITPD would have no real affect on any of these operations, since the changes are internal to the unit.

As far as lading and placard information is concerned, a new classification of car would be required complete with handling regulations. In general, there would be no major changes during transit, however the carrier should be aware that the car is fitted with an ITPD in the event of a derailment or fire.

Finally, a tank car equipped with an ITPD must be designed to withstand the loads imposed upon it by sloshing, during coupling and braking actions. The ultimate plate thickness of the inner shell will result from this analysis. This aspect was considered in previous sections.

3.3 CHANGING COMMODITIES

Many CTC-112J class tank cars are operated under dual lease, with the lading alternating between LPG and anhydrous ammonia over the course of the year. The procedure for changing these commodities is a straightforward one. However, in the case of an ITPD tank car, the existing procedure becomes inadequate.

3.3.1 Existing Procedure (Reference 12)

Assuming a rail tank car has been transporting LPG and is to be converted to anhydrous ammonia, the first step is to connect the tank to a flaring stack. This operation consists of venting the residual gas and liquid remaining in the tank after use. The rate of removal is dependent on the rate the lading can be burned in the stack. The addition of an ITPD would have no real affect on this procedure as it parallels unloading.

3.3.2 Internal Tank Cleaning

After flaring, steam is pumped into the tank for 6-8 hours to heat and clean the tank of buildup. With an ITPD, the time for the steam to heat the car would increase due to the additional mass, and because the steam circulation would be restricted to some extent.

Next, there is a period of air drying for 2-4 hours. Again, the presence of the liner would restrict air flow and extend the drying time needed. Although it appears that this procedure could still be used for an ITPD equipped tank car, tests need to be performed to determine

the minimum times for steaming and drying. These times should reflect a finished product equal to that which a CTC-112J class car would yield under the normal conditions.

3.3.3 Manway Access

The construction of the pressure fittings in a CTC 112J class tank car is such that a central manway of 460 mm (18 in.) ID is capped by a cover. The entire valve train, supplemented with thermometer and gauging device can be removed in its entirety, simply by unbolting it and lifting out the hatch from its seat on the tank housing. This procedure allows access to the tank interior for inspection or cleaning.

As a final step in the commodity change, the manway is opened and personnel entry is made to mop up condensate collected on the tank bottom and to perform a visual inspection, primarily for corrosion damage or surface cracks. This step is critical in that an unknown quantity of water is now present in the tank. It must be removed to prevent eduction pipe freeze-up during the winter months and to keep the lading to the specified maximum water content allowable.

The introduction of an ITPD presents a problem not only for internal access but for inspection purposes as well. In its basic form the device is an inner shell closed at the top and separated from the tank shell by a specific annular gap. Thus, for the eduction pipes and associated instruments to reach into the sump area, holes must be provided in the device.

The necessity of access into the tank interior requires that a hatch be included in the liner. This could be accomplished by the addition of an internal liner hatch connected directly to the eduction pipes and associated equipment (Figure 3.3.3-1). This secondary hatch would be manufactured integral with the valve train, and must house the vent valve as well. In practice, the unit would self-seal over a mating flange on the inner shell as the primary manway is bolted down. This arrangement preserves both the internal thermal protection function and allows internal access into the shell.

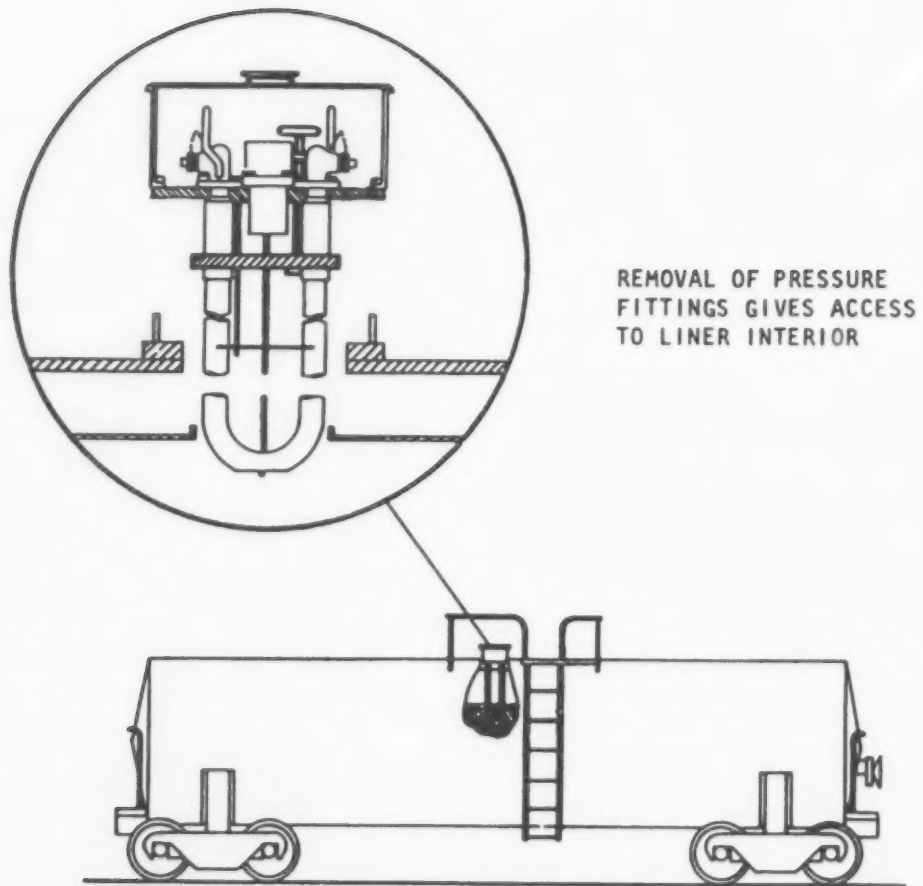


FIGURE 3.3.3-1 INTERNAL ACCESS CONCEPT

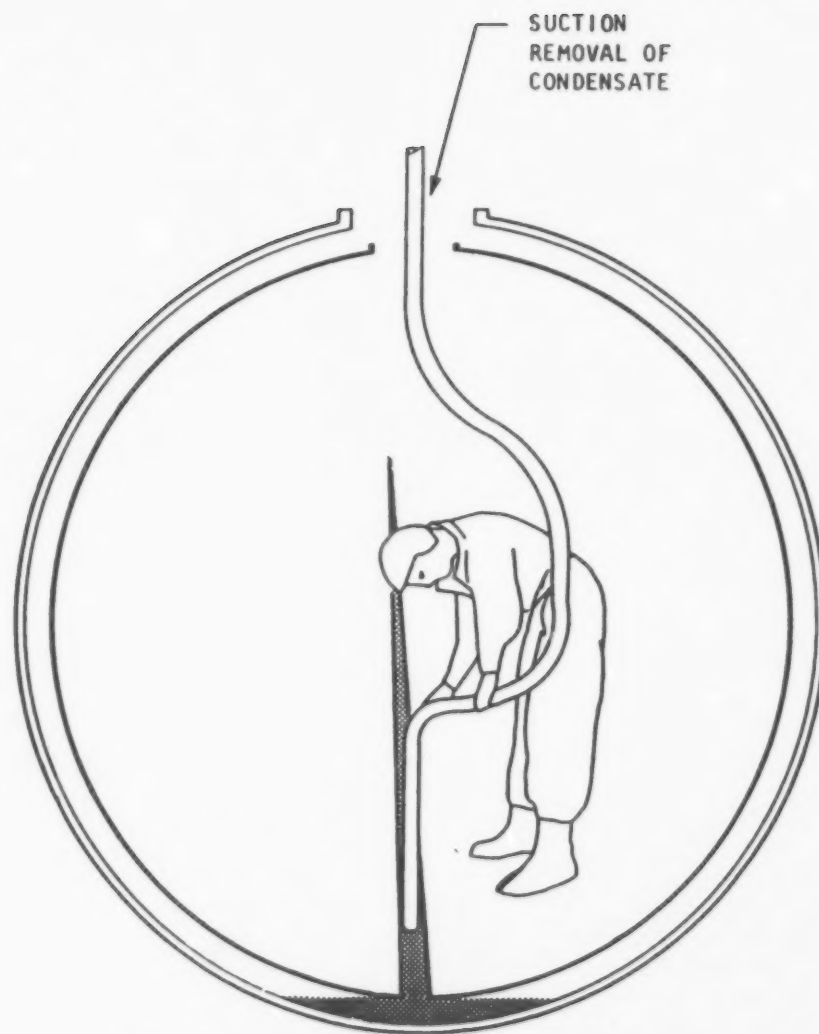


FIGURE 3.3.4-1 CROSS SECTION OF TANK CAR WITH ITPD - REMOVAL OF CONDENSATE

3.3.4 Inspection and Mop-up

Inspection and mop-up is part of the procedure for changing commodities. With the addition of the internal shell, both of these activities are affected.

As the ITPD is open at the bottom (Figure 3.3.4-1) access to the tank shell is available. The width of the gap is not well defined at this point, as it would be a result of the thermal study. However, it is estimated to be in the range of 250-500 mm (10-20 in.)

For the purposes of mop-up, the main objective is to remove condensate. With that in mind, the gap would be sufficient as virtually all the liquid would be on the tank bottom after a lengthy drying cycle.

A visual inspection of the entire tank shell cannot be done as the ITPD blocks the view. The elimination of this procedure may not be a severe drawback however. The CTC specifies that all tank cars have their relief valves tested at 5 year intervals and the tank shells inspected and pressure tested every 10 years, or after major structural repairs. There is no mandatory inspection called for during a commodity change. Further, this visual inspection is subjective at best and no details of the cars integrity is recorded, a tank merely passes or fails and the date recorded. If, however, there are critical areas that can only be inspected from inside the tank (dents, gashes, etc. can be seen from an external inspection) then access to these critical areas could be facilitated with the strategic placement of removable access ports. However, this study has not been able to identify any such critical areas based on discussions with operators.

As this type of visual check is part of the 10 year inspection it will be examined further in subsection 3.4.

3.4 PERIODIC INSPECTIONS

The CTC Regulations (13) regarding periodic inspections state that a CTC-112J class car must have its safety relief valve checked at periods of no longer than 5 years. The tank shell itself is to be tested and inspected at 10 year intervals, or after structural repairs to the pressure vessel.

The testing procedure consists of a visual check for corrosion damage, i.e., pitting, scale buildup, etc., a relief valve check, and a hydrostatic test to design

pressure for 10 minutes. The addition of an ITPD would have no real affect of the pressure test, aside from the implementation of the proper loading procedure during filling. However, the visual inspection usually performed internally is not possible due to the inner shell.

At present, the inspection procedure is an informal test requiring very little in the way of documentation. As previously stated, the car passes or fails and the date is recorded on an inspection certificate and stenciled on the unit.

This type of check is rather subjective at best and the lack of a report or documented procedural checklist makes it all the more questionable. The addition of an ITPD would raise the necessity of a non-destructive type of inspection.

Contact with a rail tank car manufacturer and lease issuer (14) revealed that virtually no unit has ever been removed from service because of what has been found during a 10 year inspection. Further talks with a design engineer showed that one major concern is weld cracks with increasing tank car age.

Methods such as mag particle, dye penetrant or x-ray analysis would identify these problems while the crack was still invisible to the naked eye.

3.4.1 Non-Destructive Testing Methods

Several methods have been investigated as alternatives to a visual inspection. These procedures are listed in Table 3.4.1-1, complete with cost estimates and descriptions.

NO.	POSSIBLE INSPECTION METHODS	COST
1	Hydrostatic Testing	\$1,500.00/tank
2	Infrared Camera Analysis	\$2,500.00-\$3,000.00/tank
3	Dye Penetrant	\$2,500.00-\$3,000.00/tank
4	Mag Particle Test	\$1.50/ft
5	Ultra Sonic Testing	Not feasible (spot check)
6	Fibre Optics	\$3,000.00 (2 days labour)
7	X-Ray Photography	Not feasible (spot check)

TABLE 3.4.1-1 NON-DESTRUCTIVE TESTING METHODS
(Reference 15)

3.4.2 Hydrostatic Testing

This procedure is a relatively simple one to perform and is at present part of the 10 year scheduled inspection for pressure cars. The test consists of filling the tank completely with water and then applying design pressure for a period of 10 minutes. During this time the tank is inspected for signs of leaking. A successful test is accomplished with a tank holding full pressure for the entire duration.

While this method does little to identify the cause of a leak it is very useful in locating questionable areas just beginning to "weep" at high pressure.

3.4.3 Infrared Camera Analysis

Infrared camera analysis is a process by which the car is filled with water at either a higher or lower temperature than ambient. The tank is then photographed with infrared cameras. The result is a series of pictures showing heat flow through the tank wall. Proper analysis would reveal areas of relative slow propagation as potential sites where crud or corrosion products have accumulated, while higher rates indicate metal thickness reduction, again probably due to corrosion.

This process would be useful to locate areas which should be visually checked. The whole process would require about 1 day duration.

3.4.4 Dye Penetrant Test

Dye penetrant tests employ the use of a marking fluid which is sprayed over the tank shell exterior. Later, the dye is washed off. During the time it was on the shell, the material shows an affinity for pits and cracks and thus makes them stand out for inspection when the surface is cleaned.

This method is less helpful for the situation of an ITPD car as no information about the interior surface is revealed. All results are confined to the outside.

3.4.5 Magnetic Particle Testing

Magnetic particle testing employs the use of a fine iron powder and electric current to present a visual pattern of

surface irregularities on a sample. While primarily a surface check, it does reveal information about the material to some depth, possibly through the entire 16 mm (5/8 in.) thick shell. The cost is relatively high however, and its use is mainly restricted to welded areas, checking for cracks and inclusions. It is not practical to inspect an entire ITPD tank car using this method.

3.4.6 Ultrasonic Testing

Ultrasonic testing is a procedure employing high frequency sound waves and an echo detector. The system finds its most practical use in thickness measurements, and for detecting cracks. However, this method is a continuous one and checks only one "spot" at a time. Thus, it is impractical to inspect the entire surface of a tank car. Further, pitting corrosion which results in deep penetration but with relatively little metal thickness reduction would not be observed with this equipment.

3.4.7 Fibre Optic Inspection

This type of inspection employs the use of a visual sensor and fibre optic technology. The result is a visual inspection on a monitor, of hard to reach areas.

In the case of an ITPD equipped tank car, direct visual information about the shell interior can be obtained. In practice, some sort of guide would be run through the annulus from the lower opening in the liner, and a fibre optic scanner run up to the area in question. This method is ideal for visual confirmation of a questionable area before deciding to cut and replace sections of the pressure vessel.

3.4.8 X-Ray Photography

X-rays offer a very good profile of a welded joint, and this is their primary use. The method is however a spot check at a relatively high cost and because of this it is not feasible for large surface areas. The use of X-ray photography is however excellent for inspection of welded patches on tank cars after structural repairs.

3.4.9 Inspection of ITPD Tank Cars

Of these non-destructive methods, no one test is sufficient to ensure complete detection of corrosion or cracks at a feasible cost. However, by using a combination of methods, a comprehensive inspection can be achieved.

Although further study is required, it seems that a good procedure would involve hydrostatic testing, infrared camera photographs followed with fibre optic analysis of questionable areas. Any repaired sections would continue to be radiographed as per CTC and AAR regulations.

3.5 REPAIR PROCEDURES

During the service life of a tank car, it may be necessary to perform repairs to fractures, pits, corroded areas or deformed and scored sections. In addition, scheduled maintenance is performed on the brake system, trucks and couplers as well as repainting and repair of supplementary systems such as access ladders and catwalks.

Different procedures are employed, depending upon the nature and extent of the damaged area (11). The various actions include chipping, drilling, grinding, welding, cleaning, patching and the performing of post weld heat treatments and radiographic examinations. Regardless of the type of activity performed however, direct access to the damaged area is required.

A tank car equipped with an ITPD would have no effect on operations completed from the exterior of the unit. In fact, if the thermal study revealed that the ITPD can safely vent a lading without a jacketed insulation then repairs and painting of the outer shell will be easier than they are now. However, for repairs to the pressure vessel itself involving removal of damaged panels, access to both sides of the area is required to perform the welding and radiographic analysis. In this case, the inner shell must be cut away in the damaged shell area and the work completed through the access hole. Following the repair, the inner shell would be welded shut from the inside. The thickness of the inner shell permits a good penetration weld from one side, and this procedure would restore the ITPD to serviceable condition. The last step is to remove residual stresses through the use of post weld heat treatment.

3.6 TANK CAR FABRICATION AND REFIT

3.6.1 Fabrication

The basic procedure used to construct a pressure vessel for a rail tank car consists of rolling steel plate, and seam welding the segments into the proper shape.

Following this, the entire tank is heat-treated at 1050-1150°F to relieve residual stresses in the welded areas. At this point two potential problems arise:

- (a) structural sagging;
- (b) scale formation.

Since the manway area is heavily reinforced, the possibility of structural deformity exists at high temperature. To prevent this, sections of 76 mm (3 in) diameter pipe are tack welded to the tank shell from the bottom of the manway area (16). This forms a support at the 6 or 8 points where the sag is expected. Later when the tank cools, the pipes are removed and the tank is ready to be hydrostatically tested.

During the time the tank is exposed to elevated temperatures, a reddish oxide film is formed on the surface of the steel. Because this film is easily removed, no sandblasting or chemical treatments are employed, the car is merely hydrostatically tested to design pressure and the flushing action of the water sufficiently cleans the tank interior. At this point, no other preparations are necessary and the car is ready to be put into service.

A car equipped with an ITPD would cause no additional problem with oxide film removal if the inner shell is constructed with the same material as the pressure vessel. Thus the existing procedure would not be altered. To ensure no sagging takes place during stress-relief it may be necessary to reposition some of the support pipes from the locations normally used.

Alternate locations should be located in line with the channel section support brackets. It is not anticipated that relocating these braces would cause any real problem, as access to support the main manway is still available, only secondary internal supports need be moved.

3.6.2 Refit

Also of significant importance in conversion to ITPD equipped tank cars is the refit of existing units. There are only two approaches possible to install the inner shell, one being internal assembly of pre-fabricated sections put in through the manway, the second method, being the insertion of the entire inner shell through the end of the tank (ie. cutting off one elliptical end cover and later re-welding it).

The first method is for all intent impractical. The manway is only 460 mm (18 in.) in diameter, and the large amount of welding required in a closed area to complete the structure is at best an extremely difficult task.

The second method is the most practical. Cutting off one elliptical cover allows the main part of the inner shell assembly to be positioned directly in the tank (Figure 3.6.2-1). In practice, the pressure vessel would be prepared by installing a number of channel section support brackets in specific locations. The inner shell would also be designed with a pattern of holes in it to match the location of the supports. Then, after assembly, the ITPD would be attached to the supports by the use of plug welds applied from inside the liner (Figure 3.6.2-2). Materials and ventilation would be supplied through the open end. This procedure would ensure a secure fit and proper annular gap around the entire structure.

The cost to install a prototype ITPD in an existing tank car would be approximately \$36,000.00. This price would include the following:

- all labour and materials to fabricate the inner shell
- all fixtures and installation labour
- all consumable materials
- any inspections, or radiographic testing of welds.

It should be noted that this price is a budgetary estimate only, and is based upon a prototype batch of three units. Subsequent production runs should significantly reduce refit costs as the quantity would justify construction of manufacturing aids such as jigs, fixtures and templates to speed up the process. A budgetary estimate of cost for installing the ITPD in a retrofit production line environment is \$25,000.00 per unit.

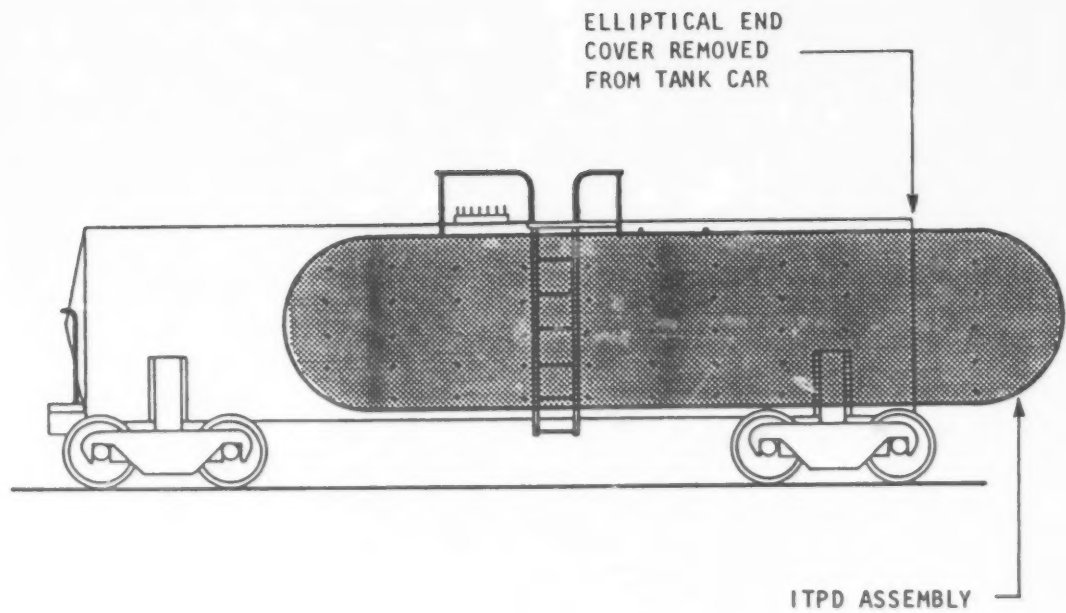


FIGURE 3.6.2-1 REFIT OF EXISTING TANK CARS

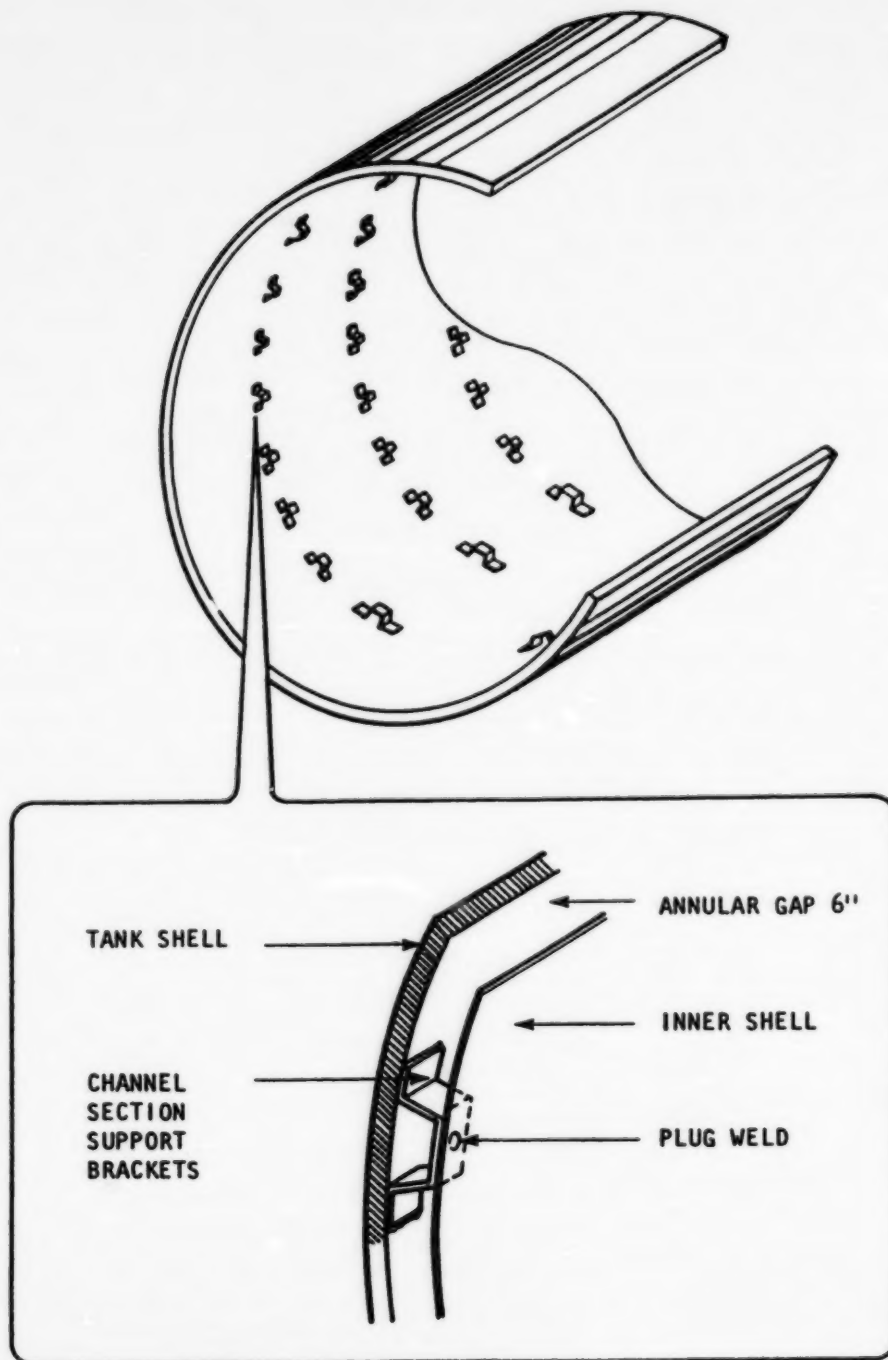


FIGURE 3.6.2-2 PROPOSED TANK/INNER SHELL ASSEMBLY

The incremental cost of fitting a new CTC-112J class tank car with an ITPD should be even lower than production run refit costs due to operational overlap. Typically, procedures such as post weld heat treatments, hydrostatic testing, and inspections are performed in any event on new cars so the addition of an ITPD at this stage does not require the additional expenditure as it does in refit operations.

3.7 CORROSION

One important consideration in determining the operational lifetime of an ITPD is the rate of corrosion when exposed to LPG or anhydrous ammonia.

Corrosion rates are expressed in many ways but the most common is "mils per year". Used as a guide, an estimate obtained from an accelerated test can be used to determine the corrosion allowance needed to give the specified life (17). In the case of a CTC-112J class tank car this is forty years.

Research into this area did not yield a specific rate, with most references merely offering trends. In all cases, the information stated that steel is quite satisfactory in an uncoated state in these two environments. Consultation with metallurgists and corrosion engineers indicated that the basic trends are formed through experiments or documented case studies and that examining the shell of older cars presently in use, would be a good indication of the corrosion rate. Contact with a rail car manufacturer (18) showed that even in cars with over twenty years in service reduction in wall thickness due to corrosion is virtually negligible.

Although it seems there need be no corrosion allowance made in the inner shell thickness, it is recommended that an accelerated corrosion test be performed to confirm this assumption before the final shell thickness is determined.

3.8 GOVERNMENT REGULATIONS

The introduction of an ITPD tank car would require the approval of the Canadian Transport Commission on its specifications prior to its construction. Several parties, including the Committee on tank cars, Mechanical

Division of the Association of American Railroads, must be consulted in order to evaluate the proposal fully. Subpart A, Section 79.4 of Reference 13 outlines briefly the requirements to justify a new specification for a class of tank car.

If the proposal illustrated the safety advantage of an ITPD tank car over more conventional jacketed insulation units, a section by section examination of the regulations governing handling of LPG and anhydrous ammonia would be in order. In particular, the following sections would be revised to accommodate the new design.

Subpart C - Specifications for pressure tank cars (classes CTC 105, 109, 112 and 114)

Sections 79.100 - General specifications applicable to pressure tank cars

79.101 - Individual specifications requirements applicable to pressure tank cars

79.102 - Special commodity requirements applicable for pressure tank cars.

Assuming the acceptance of the ITPD equipped tank car for use in Canada, a problem exists regarding refits, and border crossings of U.S. based tank cars.

4.0 CONCLUSIONS AND RECOMMENDATIONS

The feasibility of using an internal shell for thermal protection of pressurized rail tank cars has been studied from mechanical and operations viewpoints. A preliminary mechanical design for the internal thermal protection device has been presented, partly based on slosh load considerations, and operations and maintenance procedures.

Despite the limitations of the sloshing analysis, the computed results are sufficient for the present stage in the design of the internal protection device for tank cars. The predictions present general estimates of the slosh loads on the internal shell resulting from rail car decelerations in typical switchyard operations, and indeed are thought to be conservative. The pressure distributions qualitatively agree with experimental results in the literature. The computer code used in the solution has been extensively tested by its originators.

A complete and detailed sloshing calculation would require a more sophisticated and expanded computer code which would provide a fully three dimensional solution; and pertinent experimental results to validate the computations. It is recommended that an experimental program be undertaken to determine wall pressures and loads that occur due to sloshing. A three dimensional version of SOLA-VOF, which is presently being developed for public use (6), should be acquired in order to enhance the design process. The three dimensional code, HYDR-3D (7) which is a descendent of SOLA-VOF, has been developed and is now available, however at substantial cost.

Using the results of the sloshing calculations, the structural portion of the study showed that the internal shell can be incorporated into the design of the tank car. An inner shell plate thickness has been selected, and one possible support scheme is presented.

The support scheme calls for a 3.2 mm thick inner shell with supports spaced every 1.2 m along the inner shell axis, and every 30° around its circumference. Along the edge of the inner shell bottom slot, supports are required every 0.3 m. As well, reinforcing members are necessary along the slot edge, and between supports, across the slot every 0.6 m. This scheme survives all required operational load cases.

The support layout described here is probably over-designed, especially around the inner shell bottom slot, due to the conservative assumptions made regarding sloshing pressure

distributions. It is likely that if a three dimensional sloshing study were performed, the support requirements along the inner shell slot would be reduced.

Overall, the introduction of an ITPD into CTC-112J class tank cars does not create any major unresolvable problems. However, numerous minor changes are required in the way operations are presently carried out, notably in inspection and loading procedures.

Although a direct visual inspection is not possible, an adequate inspection procedure can be developed for an ITPD equipped tank car employing non-destructive methods. And even though the level of effort is increased substantially from the existing system, the information gained results in a highly documented inspection and should contribute to an overall improvement in tank car safety.

In general, the operations affected most are the more infrequent ones over the typical life of a unit. Thus, the extra downtime incurred due to more elaborate procedures is only a small percentage of the total usability of a typical tank car.

The preliminary analyses presented in this report show that the internal thermal protection device is feasible for pressurized rail tank cars. Further study is required in the areas discussed in this report in order to provide more conclusive evidence for the viability of the system, particularly in the area of sloshing. However, the anticipated results of further study are expected to substantiate the conclusions presented here. No problems are expected to arise from the structural point of view, and there do not seem to be any unresolvable problems from the operations and maintenance standpoint.

It is recommended that the project proceed to the next task covering the thermal and fluid analysis aspects. This will determine whether the overall concept is viable and whether a scale test is warranted.

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APPENDIX A

INNER SHELL FATIGUE ANALYSIS

FATIGUE ANALYSIS OF INNER SHELL

The following fatigue calculations are based on methods presented in Reference 8. For TC-128 - Grade B steel assume the following mechanical properties at room temperature:

$$S_{ut} = 85 \text{ ksi (tensile strength)}$$

$$S_y = 60 \text{ ksi (yield strength)}$$

For axial loading the mean endurance limit is given by:

$$\begin{aligned} S_e &= 19.2 + 0.314 (S_{ut}) \\ &= 46 \text{ ksi} \end{aligned}$$

The following modifying factors are applied to the endurance limit:

- i) K_a - Surface finish

Assuming inner shell material is hot rolled:

$$K_a = 0.59$$

- ii) K_b - Size effects

For axial loading K_b may be taken as 1.

- iii) K_c - Reliability

For 99% reliability:

$$K_c = 0.814$$

iv) K_d - Temperature effects

The sloshing loads occur under atmospheric conditions, and for $T < 840^{\circ}\text{F}$,

$$K_d = 1$$

v) K_e = Stress concentration factor

This factor was calculated based on Appendix B of Reference 9.

$$K_e = \frac{1}{K_T}$$

$$t = \frac{1}{8} \text{ (thickness of shell)}$$

and using $r = \frac{1}{4}$ for the fillet radius,

This gives

$$\frac{r}{t} = 2$$

Which yields

$$K_T = 1.5$$

Therefore

$$K_e = \frac{1}{1.5} = 0.67$$

Therefore the adjusted endurance limit is:

$$S_e = K_a K_b K_c K_d K_e S_e$$

$$= (.59)(1)(.814)(1)(.67)(46)$$

$$= 14.8 \text{ ksi}$$

THE S-N CURVE

At $N = 1000$ cycles assume the fatigue strength is:

$$\begin{aligned} S_f &= (.65) (S_{ut}) \\ &= 55 \text{ Ksi} \end{aligned}$$

The S-N curve is defined by:

$$\log S_f = b \log N + C$$

$$\text{and } b = -\frac{1}{3} \log \left(\frac{.65 S_{ut}}{S_e} \right)$$

$$= -\frac{1}{3} \log \left(\frac{55}{14.8} \right)$$

$$= -.1900$$

$$C = \log \left(\frac{(.65 S_{ut})^2}{S_e} \right) = \log \left(\frac{55^2}{14.8} \right)$$

$$= 2.310$$

From the finite element analysis the maximum stress in the inner shell is:

$$\sigma_{\max} = 159.2 \text{ MPa} = 23.1 \text{ Ksi}$$

The number of cycles corresponding to this stress is:

$$N = 10^{-(c/b)_S (1/b)}$$

$$= 90.1 \times 10^3 \text{ cycles}$$